Educational Administration: Theory and Practice
2024, 30(4), 8846-8853
ISSN: 2148-2403
https://kuey.net/

Research Article

"Investigation Variable Parameters in Cryocooler Design: CFD simulation of Coaxial Pulse tube Cryocooler"

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Citation: B. V. Patel, Dr. Jatin H Patel, (2024), "Investigation Variable Parameters in Cryocooler Design: CFD simulation of Coaxial Pulse tube Cryocooler", *Educational Administration: Theory and Practice*, *30*(4), 8846-8853
Doi: 10.53555/kuey.v30i4.2860

ARTICLE INFO

ABSTRACT

The most practical choice for compact and convenient pulse tube refrigeration in real-world scenarios is the inline type. To anticipate the lowest achievable temperature in the cold heat exchanger under a refrigerating load-free condition, a Computational Fluid Dynamics(CFD) analysis of the Coaxial pulse tube Cryocoolers(CPTC) is conducted. Helium serves as the working fluid for this analysis. This study meticulously examines the required phase lag between mass flow rate and temperature, while also observing temperature and density variations. The cryocoolers can achieved lowest temperature nearly 111 K with 0-watt load and 155 K with 1watt refrigerating load at cold heat exchanger with maximum enthalpy flow rate achieved when the phase angle is around 30°. if we consider axial conduction losses than we can find more accurate prediction about cooling effects.

Key Words: CFD analysis; Coaxial Pulse tube cryocoolers; cryocoolers; Frequency results.

NOMENCLATURE

C_2	Inertial resistance factor
dh	Hydraulic diameter of screen
dw	Wire diameter of screen
E	Energy (J/kg)
f	Operating frequency (Hz)
h	Enthalpy (J/kg)
k	Thermal conductivity (J/kg K)
l	Mesh distance
m	Mesh per inch
n	Number of packed screens per length
p	Pressure (Pa)
r	Radius (m)
Reh	Reynolds number
t	Time (s)
v	Velocity (m/s)
X	Amplitude
x	X axis
xt	Screen transverse pitch

Greek symbols

ω	Angular velocity (rad/s)
ρ	Density (kg/m^3)
δk	Thermal penetration (m)
ε	Porosity
μ	Viscosity (Pa s)
α	Permeability
β	Opening area ratio of screen
τ	Viscous stress

Subscripts

0	Time-averaged
f	Fluid
s	Solid

INTRODUCTION

A pulse tube Cryocooler (PTC) is distinguished by its cold section devoid of moving parts, offering the potential for increased reliability and reduced vibration, particularly when powered by a linear compressor, thus surpassing other small cryocoolers. For practical applications, the inline type stands out as the most compact and convenient pulse tube cryocoolers.Interestingly.it frequently serves as a seamless replacement for sterling cryocoolers without necessitating any modifications to the Dewar or connection to the cooled device.

The orifice pulse tube introduced by Mikulin[1] and the double intel pulse tube developed by Zhu[2] have enabled pulse tube configuration to achieve performance levels that rival, and in certain cases surpass, those of stirling cryocoolers.

One drawback of the orifice pulse tube is the requirement for the mass flow rate to consistently lead the pressure. In extreme scenarios. The mass flow rate and pressure may align in phase with each lead the mass flow rate at the inlet to the expansion space.

While the addition of the secondary orifice typically resulted in enhanced efficiencies compared to an Orifice Pulse Tube Cryocooler (OPTC). A double Inlet Orifice Pulse Tube Cryocooler's (DIOPTC) performance was not consistently reproducible. [4]

As an alternative method for adjusting the phase between pressure and mass flow rate, Kanao et al. Proposed the use of an Inertance tube. Unlike the orifice, the inertance tube dose not face the same limitations; by carefully selecting the geometry of the Inertance tube, It is feasible to control whether the mass flow rate leads or lags the pressure. Inertance refers to dynamic characteristic associated with the compliance of terminating reservoir, Inertance contributes to the complex impedance at the warm-end of the pulse tube.

COAXIAL PULSE-TUBE CRYOCOOLER

Independently, Richardson, R.N at Oxford University Engineering Laboratory proposed the design enunciated as Coaxial Pulse Tube Cryocoolers. The regenerator is housed with the annular space, while the central cylinder serves as the pulse tube, as depicted in Figure (1) in the subsequent section on CFD and Model.

CFD AND MODEL

To comprehend the primary factors influencing the performance of a Coaxial Pulse Tube Cryocooler (CPTC), commercially available Computational Fluid Dynamics (CFD) software like Fluent® proves to be a robust tool. It enables the numerical solution of the Navier-stokes equations utilizing the finite volume discretization scheme.

The thermos-fluidic processes within Pulse Tube Cryocoolers are (PTCs) are intricate, and the mechanisms governing their performance remain insufficiently understood. In this investigation, the commercial Computational Fluid Dynamics software Fluent ® was employed to model the entire Coaxial Pulse Tube Cryocooler system. The simulations depict a fully-coupled system operating in steady-periodic mode, devoid of any arbitrary assumptions. The aim was to explore the impact of two-dimensional flow effects on the performance of CPTC. Computer simulations were conducted under load free refrigerating conditions to ascertain the lowest achievable temperature and analyze the required phase lag between mass flow rate and cold end temperature.

According to the reference by Ashwin T.R[5], for both the design data of the Coaxial Pulse Tube Cryocooler and its simulation, the volume of each component remains consistent, the specific dimensions and materials of the components are outlined in Figure 1 and table 1, respectively.

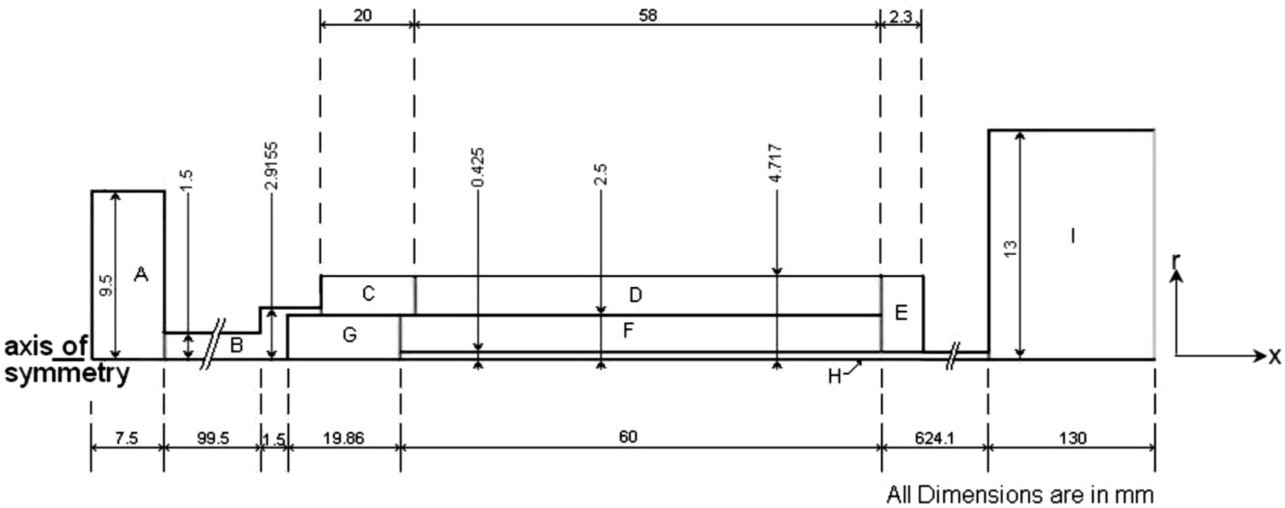


Figure 1: Schematic Diagram (CPTC)

Table 1: Different Component of CPTC

Component	Radius (Meter)	Length (Meter)	Material
A(Compressor)	9.54E-03	7.5E-03	S.S-304
B(Transfer Line)	1.55E-03	1.01E-01	S.S-304
C(WHX 1)	4.717E-03 – 2.5E-03	2E-02	OFHC-Copper
D(Regenerator)	4.717E-03 – 2.5E-03	5.8E-02	S.S-304
E(CHX)	4.717E-03 – 4.25E-04	2.3E-03	OFHC-Copper
F(Pulse tube)	4.717E-03 – 4.25E-04	6E-02	S.S-304
G(WHX 2)	4.00E-03	1.986E-02	OFHC-Copper
H(Inertance tube)	4.25E-04	0.6841	S.S-304
I(Surge Volume)	0.013	0.13	S.S-304

CFD SIMULATION PROCEDURE

The Fluent software represents a cutting-edge tool used for simulating heat transfer and fluid flow in intricate engineering scenarios. Its versatility allows for meshing complex geometries and addressing both 2-Dimensional and 3-Dimensional challenges. Fluent is capable of modeling transient flow, transport phenomena in a porous media, two-phase flow, and volumetrically-generating sources. It achieves this by numerically solving the continuum fluid and energy equations without relying on arbitrary assumptions.

The mesh model for the Coaxial Pulse Tube Cryocooler (CPTC) is constructed using ANSYS® Workbench® Modeler®. Various boundary conditions, including mass flow rate, pressure, axis-symmetry, and wall boundaries for all components, are applied to the mesh model, as illustrated in Figure 2.

The mesh model is exported to ANSYS® Workbench® CFD Fluent® for simulation. The Coaxial Pulse Tube Cryocooler (CPTC) is simulated using cylindrical and linear alignment, axis-symmetric, two-dimensional flow, and a laminar model based on physical velocity, with helium as the ideal gas for the working fluid. Copper is utilized for Warm Heat Exchangers 1 (WHX1), Warm Heat Exchangers 2 (WHX2), and the Cold Heat Exchanger (CHX), while stainless steel 304 is employed in all other components, including the regenerator. Thermal conductivity, specific heat, and viscosity are treated as temperature-dependent properties. The boundary conditions utilized for the simulated model are derived from the reference of J Cha et al. 2004[7], and are detailed in Table 2.

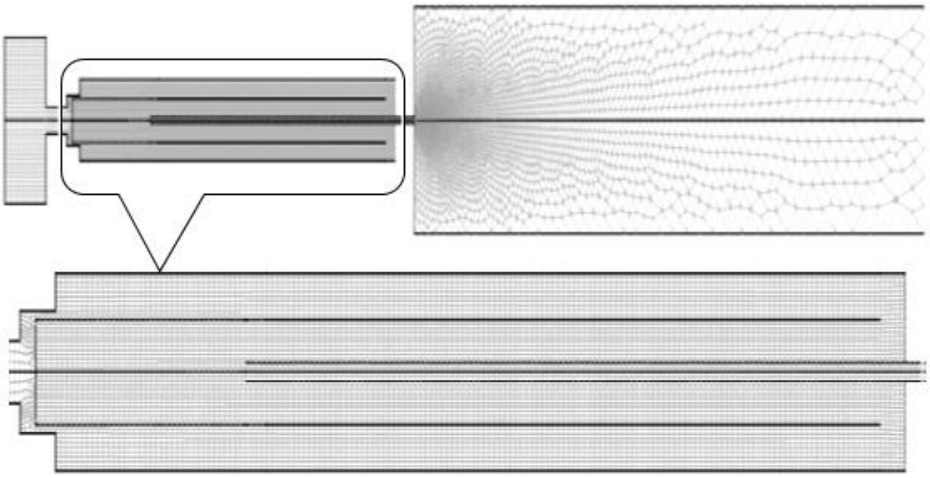


Figure 2: Two Dimensional Axis Symmetry Mesh Model of CPTC

Table 2: Boundary condition for different component

Sr. no.	**Component**	**Boundary condition**
A	Compressor wall	Adiabatic
B	Transfer line wall	Adiabatic
C	WHX1 wall	293K
D	Regenerator wall	Adiabatic
E	CHX wall	Adiabatic
F	Pulse tube wall	Adiabatic
G	WHX2 wall	293K
H	Inertance tube wall	Adiabatic
I	Surge volume wall	Adiabatic
Permeability of porous component β (m^2)		1.06e-10
Inertial resistance C(m^{-1})		76090
Initial condition		300K
CHX load		0W

GOVERNING EQUATION

The governing equations (mass flow rare equations, momentum and energy equations) solved by fluent are as follows:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r) + \frac{\partial}{\partial X}(\rho v_X) = S_m \qquad \text{(Eq. 1)}$$

Momentum Equation:

$$\frac{\partial}{\partial t}(\rho v_r) + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_r) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r v_r) = -\frac{\partial p}{\partial r} + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(2\frac{\partial v_r}{\partial x} + \frac{\partial v_x}{\partial r}\right)\right] + \frac{1}{r}\frac{\partial}{\partial r}\left[ru\left(2\frac{\partial v_r}{\partial r} - \frac{2}{3}(\nabla \cdot \vec{v})\right)\right] - 2\mu\frac{v_r}{r^2} + \frac{2}{3}\frac{\mu}{r}(\nabla \cdot \vec{v}) \qquad \text{(Eq. 2)}$$

where,
r = radial coordinate
x = axial coordinate
Vr= Radial velocity
Vx= Axial velocity

The body forces (gravity forces) and any other external forces have been neglected in the above equations.

Energy equation:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot \left(\vec{v}(\rho E + p)\right) = \nabla \cdot \left(\hat{k}_f \nabla T + (\tau \cdot \vec{v})\right) \quad \text{(Eq. 3)}$$

Where

$$E = h - \frac{P}{\rho} + \frac{v^2}{2} \quad \text{(Eq. 4)}$$

All properties correspond to the properties of the working fluid helium. The aforementioned equations are applicable to all components except for WHX1, CHX, WHX2, and the regenerator. These four components are modeled as porous zones, under the assumption of local thermodynamic equilibrium between the fluid and solid structure within them. The mass momentum and energy equations within these four components are as follows:

$$\frac{\partial(\phi\rho)}{\partial t} + \frac{1}{r}\frac{\partial}{\partial r}\left(\phi r \rho V_r\right) + \frac{\partial}{\partial x}\left(\phi \rho V_x\right) = 0 \quad \text{(Eq. 5)}$$

$$\frac{\partial}{\partial t}(\emptyset \rho v_x) + \frac{1}{r}\frac{\partial}{\partial x}(\emptyset r \rho v_x v_x) + \frac{1}{r}\frac{\partial}{\partial r}(\emptyset r \rho v_r v_x) = -\frac{\partial \emptyset p}{\partial x} + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(2\frac{\partial \emptyset v_x}{\partial x} - \frac{2}{3}(\nabla \cdot \emptyset \vec{v})\right)\right] + \frac{1}{r}\frac{\partial}{\partial r}\left[ru\left(\frac{\partial \emptyset v_x}{\partial r} + \frac{\partial \emptyset v_r}{\partial x}\right)\right] + F_{porous_x} \quad \text{(Eq. 6)}$$

$$\frac{\partial}{\partial t}\left(\emptyset \rho_f E_f + (1-\emptyset)\rho_s E_s\right) + \nabla \cdot \left[\vec{v}\left(\rho_f E_f + P\right)\right] = \nabla \cdot \left[\hat{k} \nabla T + (\tau \cdot \vec{v})\right] \quad \text{(Eq. 7)}$$

Where $\varphi = 0.69$, $\beta = 1.06\times10{-}10$ [m2] and C = 76090 [m-1] were assumed.

These parameters are based on experiments of Harvey [6] for axial flow through a randomly packed stack of 325 mesh stainless steel screens. Initially, we aimed to replicate the results of a Pulse Tube Cryocooler (PTC) simulated in existing literature. We utilized dimensions and boundary conditions provided in recent research papers for this purpose. Firstly, we generated the mesh model of the PTC and imported it into Fluent. To prepare the simulation model, we assigned solid materials and fluid properties, considering temperature dependence. Boundary conditions on the walls and fluid zones were applied according to the literature. Additionally, initial conditions were set, with mass flux as input and pressure as output. The simulation was conducted under two-dimensional, axis-symmetric, and unsteady state conditions.

The simulation yielded the desired results, enabling qualitative performance prediction through CFD analysis. The obtained results are discussed below.

RESULTS

To investigate the thermo-fluidic behavior and assess the optimal parameter range, various cases for Coaxial Pulse Tube Cryocooler (CPTC) simulation were established. The simulation setup remained consistent with the procedure discussed earlier, with the exception of changes to relevant parameters as outlined in Table 3.

Table 3: Change Parameters in CPTC

Sr. no.	Length		CHX Wall Boundary condition	Frequency input at comp. (Hz)
	Regenerator	Pulse Tube		
Case1	58	60	Heat load (0Watt)	34
Case 2	58	60	Heat load (1Watt)	34

CASE 1 (CPTC- ADIABATIC (0 WATT) - FREQUENCY-34 HZ)

Case 1 employed an adiabatic wall boundary condition at the Cold Heat Exchanger (CHX), simulating the absence of any refrigeration load applied to the entire system. A multi-dimensional model of a Coaxial Pulse Tube Cryocooler (CPTC) was simulated using Computational Fluid Dynamics (CFD), comprising 10,376 nodes. All boundary conditions closely resembled those of a real system, as outlined in Table 2, and results were obtained. Figure 3 illustrates the variation of the cold tip temperature over time. Notably, the system reached a steady-periodic state approximately 250 seconds into the simulation, with the final CHX temperature stabilizing at 111 K.

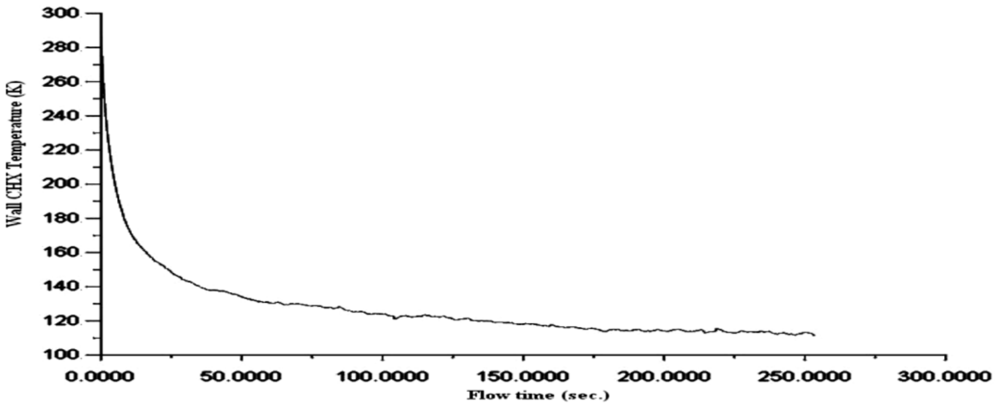
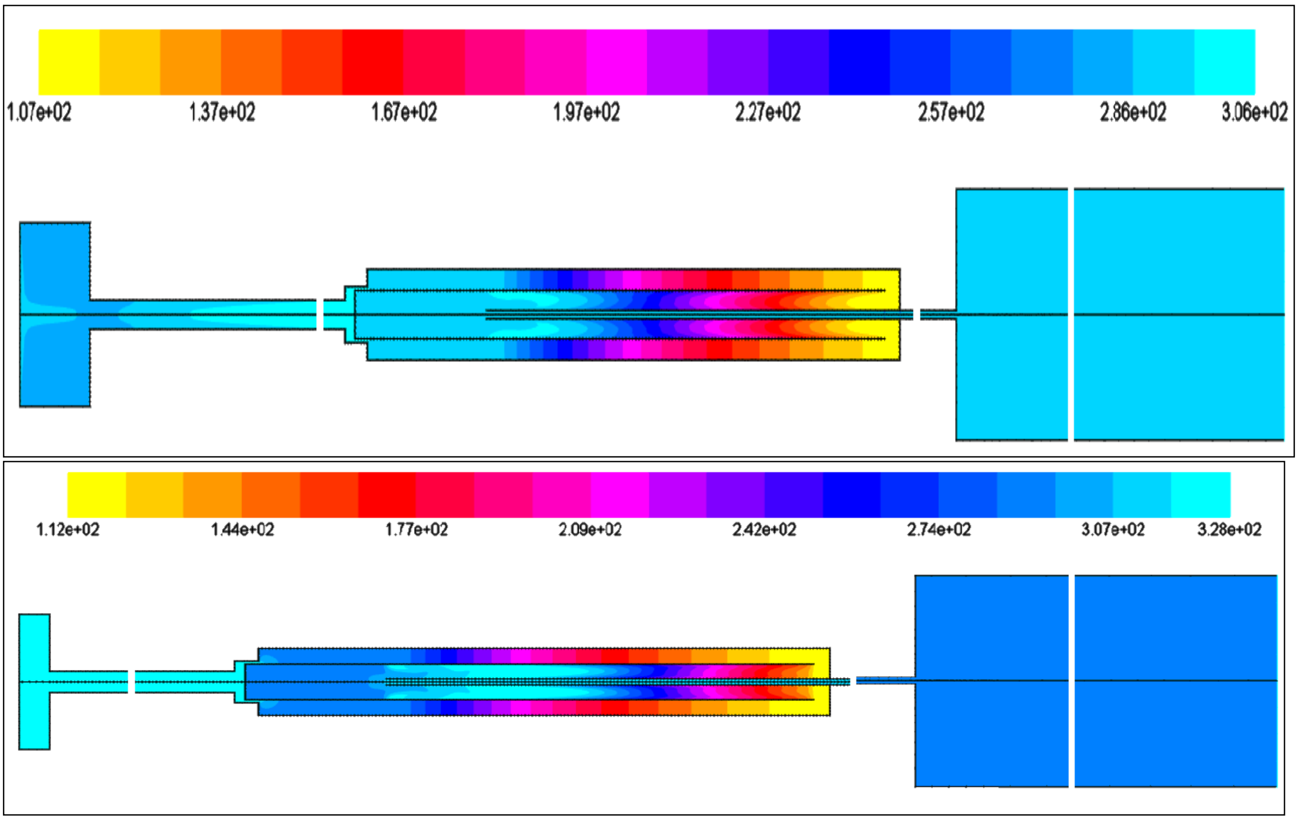


Figure 3: Case 1 -Cyclic Average CHX Wall Temperature Drop Profile

As depicted in Figure 4, the lowest temperature attained at the Cold Heat Exchanger (CHX) wall under no refrigerating load conditions is 111.3K at 250 seconds, with the temperature continuing to decrease. The system's lowest temperature, approximately 98.7 K, is observed in the right section of the pulse tube, coinciding with the region where density reaches its maximum value (12.44 kg/m³) as shown in Figure 5. Conversely, the highest temperature achieved is around 328 K at Warm Heat Exchanger 2 (WHX2). The gas within the pulse tube exhibits a segmented behavior, with a central portion remaining confined within the tube and oscillating solely around the center, akin to the displacer in a Stirling Cryocooler. As the temperature decreases at the Cold Heat Exchanger, the gas flow near the cold end becomes more sluggish due to the higher gas density. Nevertheless, fine-tuning of the phase-shifting mechanism, oscillation frequency of the gas at the inlet, and pulse tube size are found to be essential for optimal performance.

In Figure 6, the phase lag observed between the mass flow rate and temperature at the Cold Heat Exchanger (CHX) is depicted. Specifically, at the CHX, the mass flow rate was leading the temperature. It is crucial to note that the phase angle should ideally be less than 90 degrees for optimal performance. This configuration ensures that maximum enthalpy flow is achieved, typically occurring around 28o.



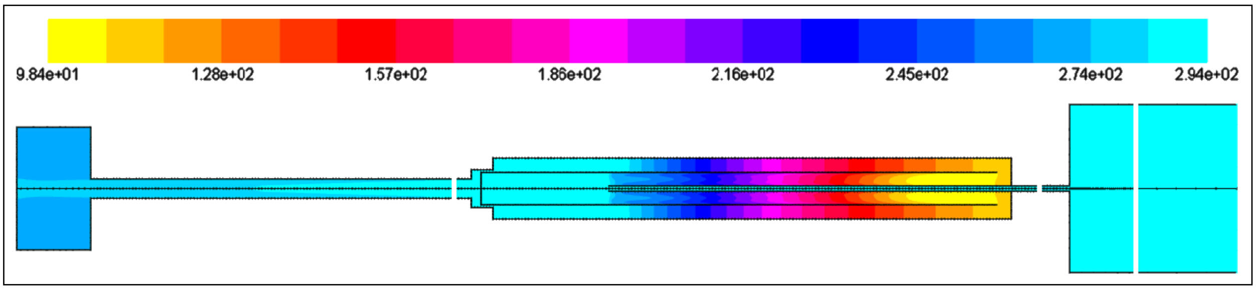


Figure 4: Case 1-Temperature Contour of CPTC.
(Temperature in Kelvin)

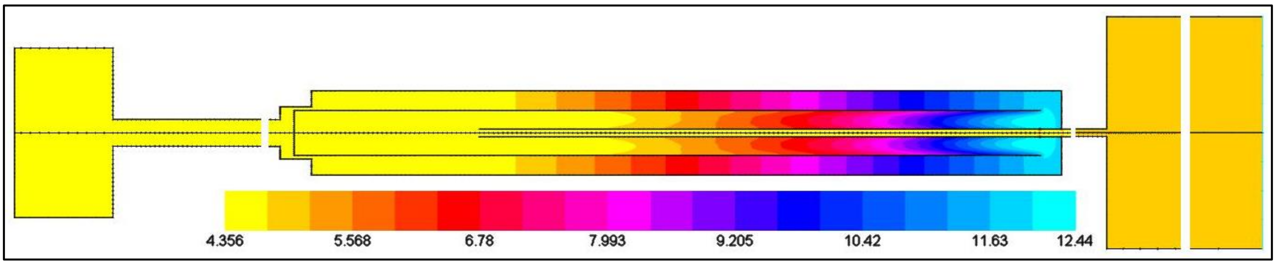


Figure 5: Case 1-Density Contour of CPTC.
(Density in Kg/m3)

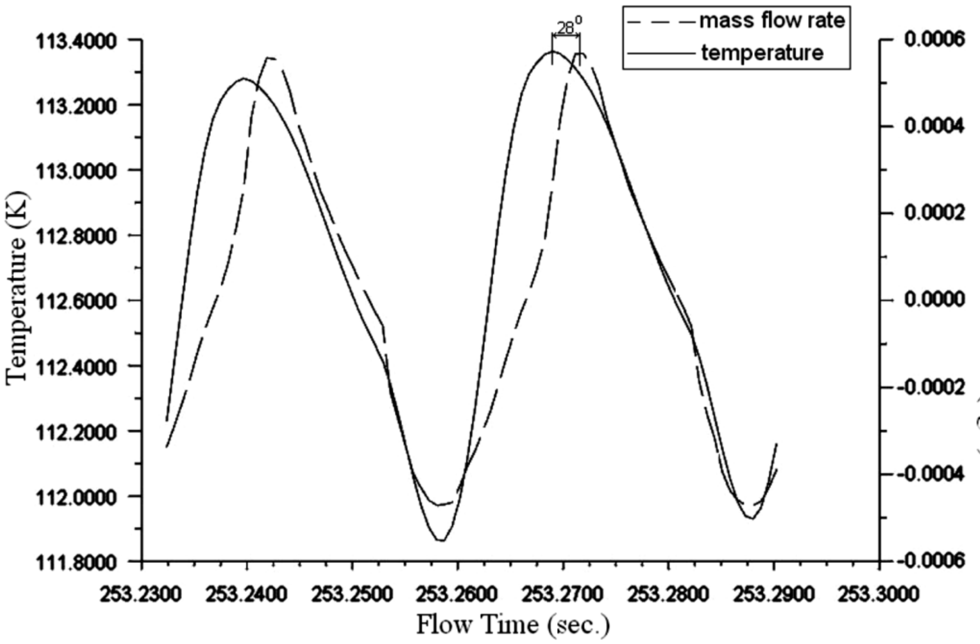


Figure 6: Case 1-Phase Lag at CHX Between Temperature and Mass Flow Rate.

CASE 2 CPTC- LOAD 1 WATT WITH FREQUENCY-34 HZ

This case represents a scenario where a constant wall heat flux is assumed for the Cold Heat Exchanger (CHX). This assumption is akin to an actual system undergoing a refrigeration process with a refrigeration load of 1W.

Figure 7 illustrates the cycle-average temperature of the surface wall of the Cold Heat Exchanger (CHX). According to the simulation, an average temperature of 157 K is predicted, indicating that a 1-watt refrigeration load occurs at this temperature. These results are obtained at approximately 100 seconds of flow time. In comparison with Case 2, which represents the same physical system and boundary conditions but with zero cooling load, the CHX has stabilized to a significantly higher average temperature. This outcome is expected because when a refrigeration load is applied to the system, the operating cold tip temperature should increase.

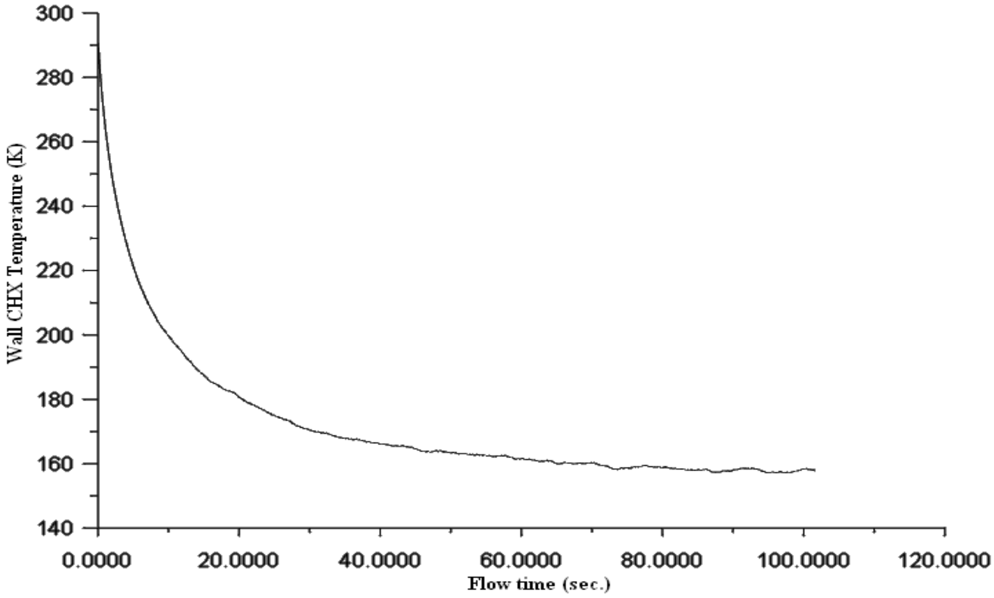


Figure 7: Case 2-Cyclic Average of CHX Wall Temperature Drop Profile.

Contours of temperature and density are depicted in Figures 8 and 9, respectively. These contours exhibit qualitative similarities to those observed in the no-load case. The maximum density value, reaching 10.94 kg/m³, is observed at the Cold Heat Exchanger (CHX), coinciding with the recorded minimum temperature.

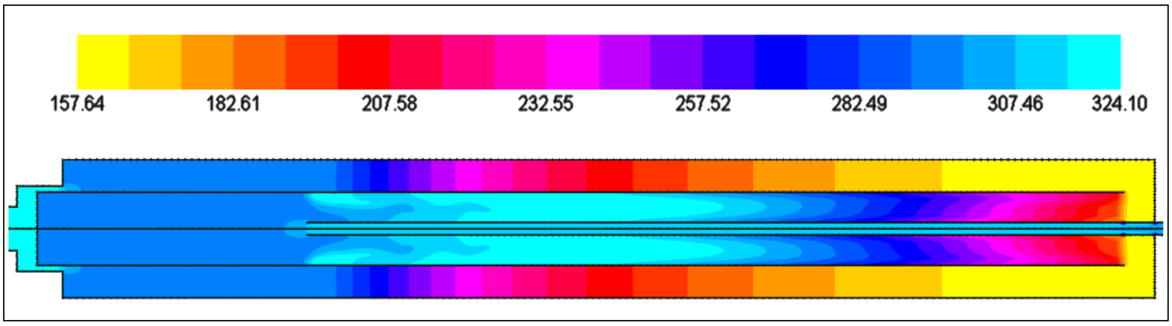


Figure 8: Case 2-Temperature Contour of CPTC
(Temperature in Kelvin)

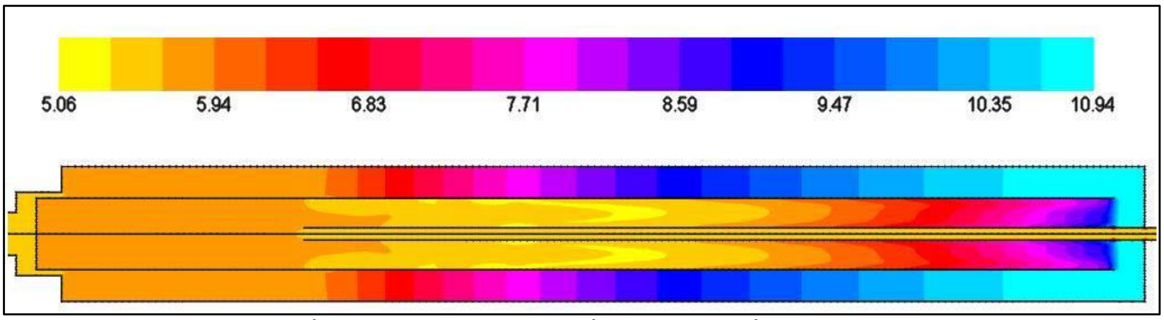


Figure 9: Case 2-Density Contour in CPTC.
(Density in Kg/m³)

CONCLUSIONS

Qualitative performance prediction can be possible with the CFD analysis. The complex behavior of thermo-fluidic process in CPTC models and detail mechanisms and their performance can be easily understanding with fluent. Piece-wise Polynomial form curve fit property gives accurate result. We can find lowest temperature111.3K with 0watt load and 157 K with 1watt refrigerating load at cold heat exchanger. The phase angle must be less than 90° for the best performance in terms of maximum enthalpy flow were achieved around 28° But axial conduction losses are not taken into account which may reduce the drop in temperature.

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