Flow Boiling and Condensation Experiment (FBCE) for the International Space Station

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Examples of Systems Demanding Predictive Models of Effects of Gravity on Two-Phase Flow and Heat Transfer

- Space Vehicle
- Astronaut Suit
- Martian Base
- Asteroid Landing
- Earth Orbiting Station
- Earth Orbiting Vehicles
- Satellites

- μ gₑ
- Moon 0.17 gₑ
- Mars 0.38 gₑ
- 1 gₑ
- 10 gₑ

Boiling and Two-Phase Flow Laboratory (BTPFL)
NASA-Supported Facilities at Boiling & Two-Phase Flow Laboratory

- High-Capacity Condensation Facility
- One-g Flow Boiling Facility
- Parabolic Flight Flow Boiling Facility
- Mini/micro-channel Condensation Facility
- Falling-Film Heating/Evaporation Facility
- Parabolic Flight Condensation Facility
The proposed research aims to develop an integrated two-phase flow boiling/condensation facility for the International Space Station (ISS) to serve as primary platform for obtaining two-phase flow and heat transfer data in microgravity.

Overriding objectives are to:

1. Obtain flow boiling database in long-duration microgravity environment

2. Obtain flow condensation database in long-duration microgravity environment

3. Develop experimentally validated, mechanistic model for microgravity flow boiling critical heat flux (CHF) and dimensionless criteria to predict minimum flow velocity required to ensure gravity-independent CHF

4. Develop experimentally validated, mechanistic model for microgravity annular condensation and dimensionless criteria to predict minimum flow velocity required to ensure gravity-independent annular condensation; also develop correlations for other condensation regimes in microgravity
Consists of:
- nPFH sub-loop
- Water sub-loop

Contains three test modules:
- Flow Boiling Module (FBM)
- Condensation Module CM-HT for heat transfer measurements
- Condensation Module CM-FV for flow visualization
1. Both One $g_e$ and Parabolic flight flow boiling experiments using single-sided and double-sided heat walls

2. Modeling of CHF for single-sided and double-sided heated walls at different orientations in Earth gravity and in microgravity

3. Computational modeling of condensing film
Flow Boiling Facility

Flow Boiling Module

Reservoir

1 atm

Liquid to Air Heat Exchanger

Air

Flow Boiling Module

Variac Watt-meter

Control Valve

High-Speed Camera

Pump

Control Valve

Filter

Turbine Flow Meter

Preheater

Watt-meter

Control Valve

T

T

P

P

T

Flow Boiling Facility

FBM H1 Thermocouples

FBM Heater Temperature Controller

FBM Heater Power Supply & Meters

FBM Power Supply

FBM Heater Temperature Controller

FBM Inlet

FBM Outlet

LED Light Source

Hard Drives for Flow Viz

FBM Inlet

FBM Outlet

FBM H1 Thermocouples

FBM Heater Power Supply

FBM Heater Power Supply & Meters

Camera Mirror

Computer for Flow Viz

Camera Data Acquisition System

Flow Boiling Module

FBM Heater Temperature Controller

FBM Inlet

FBM Outlet

FBM Power Supply

FBM Power Supply & Meters

LED Power Supply

Preheater

Pump

Control Valve

Filter

Turbine Flow Meter

ASGSR 2015  Boiling and Two-Phase Flow Laboratory (BTPFL)  November 2015
Flow Boiling Module

Flow Straightener

Support Plates (Aluminum)

FC-72 Inlet

O-rings

Outer Channel Plates (Lexan)

Flow Straightener

O-rings

Copper Slabs with Resistive Heaters

Channel Side Wall Plate (Lexan)

FC-72 Outlet

Development Length:

Heated Length:

Exit Length:

L_d = 327.9 mm
L_h = 114.6 mm
L_e = 60.9 mm

Channel Height: H = 5.0 mm
Channel Width: W = 2.5 mm

Flow Boiling Module

Boiling and Two-Phase Flow Laboratory (BTPFL)

November 2015
Heated Wall Design

188 Ohm Resistive Layer (Covered with Glass Passivation)
Solder Layer (96% Tin - 4% Gold Metallization)

Al₂O₃ Substrate
Solder Pads

1.04 mm
0.9 mm
16.4 mm
4.5 mm
0.56 mm

Oxygen-Free Copper Slab

All dimensions in mm

Heated Wall Design

Heated Wall 2: H₂
Heated Wall 1: H₁

17.3 mm
4 × 17.3 = 69.2 mm
114.6 mm
Tₜ₂₄
Tₜ₂₃
Tₜ₂₂
Tₜ₂₁

17.3 mm
Tₜ₁₇
Tₜ₁₆
Tₜ₁₅
Tₜ₁₄

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Horizontal Flow Boiling – Heated Wall Configurations

**Top wall heating**

- Flow direction: \( g_e \)
- Configuration: H1: \( q''_w \)
- Vapor Layer
- Liquid

**Double-sided heating**

- Flow direction: \( g_e \)
- Configuration: H1: \( q''_w \)
- Vapor Layer 1
- Liquid
- H - \( \delta_1 \) - \( \delta_2 \)
- Configuration: H2: \( q''_w \)
- Vapor Layer 2
- Liquid
- H - \( \delta_1 \) - \( \delta_2 \)

**Bottom wall heating**

- Flow direction: \( g_e \)
- Configuration: H1: \( q''_w \)
- Vapor Layer
- Liquid
- H - \( \delta_1 \) - \( \delta_2 \)
- Configuration: H2: \( q''_w \)
- Vapor Layer
- Liquid
Slightly subcooled flow at low velocity

$G = 394.8 \text{ to } 403.4 \text{ kg/m}^2\text{s (U = 0.25 m/s)}$

$\Delta T_{\text{sub,in}} = 3.6 - 5.1^\circ C$

- **Top Wall Heated**
  - 51%
  - 87%
  - CHF (9.3 W/cm²)

- **Top & Bottom Walls Heated**
  - 59%
  - 86%
  - CHF (18 W/cm²)

- **Bottom Wall Heated**
  - 51%
  - 85%
  - CHF (28.2 W/cm²)
Horizontal Flow Boiling – Experimental CHF Results

- \( \Delta T_{\text{sub,in}} = 24.5 - 31.0^\circ \text{C} \)
- \( \Delta T_{\text{sub,in}} = 3.3 - 7.7^\circ \text{C} \)
- \( x_{e,\text{in}} = 0.03 - 0.18 \)

Graph showing CHF [W/cm²] vs. \( G \) [kg/m²s] with data points for Top heating, Double-sided heating, and Bottom heating.
Mass Balance:

Vapor layer 1:

\[ U_{g1} = \frac{q'' \delta_0}{\rho_g \delta_1 (c_{p,g} \Delta T_{\text{sub,in}} + h_{fg})} \]

Vapor layer 2:

\[ U_{g2} = \frac{q'' \delta_0}{\rho_g \delta_2 (c_{p,g} \Delta T_{\text{sub,in}} + h_{fg})} \]

Liquid layer:

\[ U_f = \frac{U H}{H - \delta_1 - \delta_2} - \frac{2 q'' \delta_0}{\rho_f (H - \delta_1 - \delta_2) (c_{p,f} \Delta T_{\text{sub,in}} + h_{fg})} \]

Momentum Balance:

Heated wall vapor layer 1:

\[ G^2 \frac{d}{dz} \left[ \frac{x_1^2}{\rho_g \alpha_1} \right] = -\alpha_1 \frac{dp}{dz} - \frac{\tau_{w,g1} P_{w,g1}}{A} \pm \frac{\tau_{il} P_{il}}{A} - \rho_g \alpha_1 g \sin \theta \]

Central Liquid layer:

\[ G^2 \frac{d}{dz} \left[ \frac{(1 - x_1 - x_2)^2}{\rho_f (1 - \alpha_1 - \alpha_2)} \right] = -(1 - \alpha_1 - \alpha_2) \frac{dp}{dz} - \frac{\tau_{w,f} P_{w,f}}{A} \pm \frac{\tau_{il} P_{il}}{A} \pm \frac{\tau_{i2} P_{i2}}{A} - \rho_f (1 - \alpha_1 - \alpha_2) g \sin \theta \]

Heated wall vapor layer 2:

\[ G^2 \frac{d}{dz} \left[ \frac{x_2^2}{\rho_g \alpha_2} \right] = -\alpha_2 \frac{dp}{dz} - \frac{\tau_{w,g2} P_{w,g2}}{A} \pm \frac{\tau_{i2} P_{i2}}{A} - \rho_g \alpha_2 g \sin \theta \]

Energy Balance:

\[ \frac{dx_1}{dz} = \frac{dx_2}{dz} = \frac{q'' W}{\dot{m} (c_{p,f} \Delta T_{\text{sub,in}} + h_{fg})} \]
Use separated flow model to determine axial variations of:
- $U_{g1}, U_{g2}$ Near-wall vapor layer velocities
- $U_f$ Liquid layer velocity
- $\delta_f, \delta_g$ Near-wall vapor layer thicknesses

### Critical Interfacial Wavelength

$$k_c = \frac{2\pi}{\lambda_c} = \frac{\rho_f \rho_g'' (U_g - U_f)}{2 \sigma (\rho_f'' + \rho_g'')} + \left[ \frac{\rho_f'' (U_g - U_f)^2}{2 \sigma (\rho_f'' + \rho_g'')} \right]^{\frac{1}{2}} + \frac{(\rho_f - \rho_g) g_n}{\sigma}$$

where $\rho_f'' = \rho_f \coth \left(2\pi H_f / \lambda_c\right)$ and $\rho_g'' = \rho_g \coth \left(2\pi H_g / \lambda_c\right)$

**Earth Gravity:**
- $g_{n1} = g_e \cos \theta$ and $g_{n2} = g_e \cos (\theta + \pi) = -g_e \cos \theta$

**Microgravity:**
- $g_{n1} = g_{n2} = \mu g_e \equiv 0$

$$\lambda_c = \frac{2 \pi \sigma (\rho_f'' + \rho_g'')}{\rho_f'' (U_g - U_f)^2}$$

### Mean Pressure Difference Across Wetting Front

$$\bar{p}_f - \bar{p}_g = \frac{4 \pi \sigma \delta}{b \lambda_c^2} \sin(b \pi)$$

where $b = 0.20$ is ratio of wetting front length to wavelength

### Interfacial Lift-off Criterion

$$\frac{\bar{p}_f - \bar{p}_g}{\rho_g} = \rho_g \left[ \frac{q_w''}{\rho_g h_{fg}} \right]^2$$

### Surface Energy Balance

$$q_m'' = b q_w''$$

### Critical Heat Flux

$$q_m'' = \rho_g \left( c_{p,f} \Delta T_{sub,in} + h_{fg} \right) \frac{4 \pi \sigma b \sin(b \pi)}{\rho_g} \left[ \frac{\delta^{1/2}}{\lambda_c^2} \right]^{1/2}$$
CHF Model Predictions

One $g_e$ Horizontal Flow

- **FC-72**
  - $p_{in} = 99.3 - 161.8$ kPa
  - $\Delta T_{sub, in} = 1.9 - 8.4^\circ C$

**CHF Model Predictions**

- Top heating (MAE = 27.4%)
- Double-sided heating (MAE = 8.1%)
- Bottom heating (MAE = 5.6%)

One $g_e$ Vertical Upflow

- **FC-72**
  - $p_{out} = 118.2 - 148.3$ kPa
  - $\Delta T_{sub, in} = 2.8 - 8.1^\circ C$

**Microgravity & One $g_e$ Vertical Upflow**

- $g_e$
- $\mu g_e$

**Heating Data**

- Double-Sided Heating Data
  - $H_1$: 5.0%
  - $H_2$: 5.7%

**Bottom heating (MAE = 5.6%)**

**Top heating (MAE = 27.4%)**

**Double-sided heating (MAE = 8.1%)**
Single-sided versus Double-sided Heating in Earth Gravity

**Single-sided Heating**

- **135°**
- **225°**
- **315°**
- **180°**
- **270°**

- **θ = 90°**
- **Upward-facing heater**
- **Downward-facing heater**

**Double-sided Heating**

- **135°**
- **225°**
- **315°**
- **180°**
- **270°**

- **θ = 90°**
- **Upward-facing heater**
- **Downward-facing heater**

**Legend:**
- Red: Upward-facing heater
- Yellow: Downward-facing heater

**Flow Directions:**
- **Upflow**
- **Downflow**

**Gravity:**
- $g_e$
CHF Predictions for Single-sided versus Double-sided Heating in Earth Gravity

**Single-sided Heating**

- **CHF (W/cm²)**
- Angles: 0°, 30°, 60°, 90°, 120°, 150°, 180°, 210°, 240°, 270°, 300°, 330°, 360°
- Varying CHF values for different angles and flow rates

**Double-sided Heating**

- **CHF (W/cm²)**
- Angles: 0°, 30°, 60°, 90°, 120°, 150°, 180°, 210°, 240°, 270°, 300°, 330°, 360°
- Varying CHF values for different angles and flow rates

- **FC-72**
- **p_in = 100 kPa**
- **ΔT_sub,in = 3°C**

- **Heater H_a**
- **Upward-facing Heater**
- **Downward-facing Heater**

Flow rates: 0.5 m/s, 1.0 m/s, 1.5 m/s, 2.0 m/s

**Boiling and Two-Phase Flow Laboratory (BTPFL)**

ASGSR 2015 November 2015
Data Sharing Plans
- NASA Office of **Physical Science Informatics (PSI)** tasked with organizing and distributing databases to researchers in the field

- Large databases (terabytes of data) will be generated from FBCE ISS experiment

- Purdue-Glenn team will create organization structure for FBCE databases to be provided to (PSI)

- Purdue is presently exploring most effective means for packaging data for ease of use by other researchers using recent FBM Earth’s gravity data as example
Organizational Structure of FBM Database

Folder Name: Summer 2015 FBM Testing

Organization Documents
Publications, Presentation, and Summaries
Data Folders
Data Folders

Contain four filetypes:

- **Text Files** containing raw sensor data output by data acquisition system
- **Matlab Scripts** for processing raw data
- **Excel Spreadsheets** containing all relevant parameters (e.g., pressure drop, heat transfer coefficient, CHF) output by processing script
- **Image Files** for flow visualization

With subfolders used to group data by operating conditions

Full description of file paths and data file structures found in “Organizational Documents” folder
Computational Modeling
Objectives

- Study flow condensation using CFD solver Fluent
- Select an appropriate phase change model
- Study heat transfer and fluid flow characteristics over a broad range of Reynolds numbers
- Lay foundation for future computational modeling of complicated flow boiling processes
Numerical Approaches to Modeling Two-Phase Systems

• **Lagrangian**
  - Smoothed-Particle Hydrodynamics (SPH) Method: Gingold & Monaghan (1977), Lucy (1977)

• **Eulerian**
  - Level-Set Method (LSM): Osher & Sethian (1955)

• **Eulerian-Lagrangian**
**Phase Change Models**

- **Rankine-Hugoniot jump condition**
  

  \[ q'' = -k_{\text{eff}} \nabla T_i \times \bar{n} = \dot{m}'' h_{fg} \]

  \[ S_g = -S_f = \dot{m}'' \left| \nabla \alpha_g \right| = \frac{k_{\text{eff}} \left( \nabla \alpha \times \nabla T \right)}{h_{fg}} \]

  where \( k_{\text{eff}} = \frac{k_g + k_f}{g + f} \)

- **Schrage Model (1953)**
  
  Kartuzova & Kassemi (2011), Magnini et al. (2013) …

  \[ S_g = -S_f = \dot{m}'' \left| \nabla \alpha_g \right| \]

  where \( \dot{m}'' = \frac{2}{2 - \gamma_c} \sqrt{\frac{M}{2\pi R}} \left[ \gamma_c \frac{P_g}{\sqrt{T_g}} - \gamma_e \frac{P_f}{\sqrt{T_f}} \right] \)

- **Lee Model (1980)**
  
  Wu et al. (2007). Yang et al. (2008), Fang et al. (2010) …

  \[ S_g = -S_f = r_i \frac{\left( T - T_{\text{sat}} \right)}{T_{\text{sat}}} \]
  
  for condensation \((T < T_{\text{sat}})\)

  \[ S_g = -S_f = r_i \frac{\left( T - T_{\text{sat}} \right)}{T_{\text{sat}}} \]
  
  for evaporation \((T > T_{\text{sat}})\)
### Phase Change Models

#### Rankine-Hugoniot jump condition


**Pros:**
1. Ease of implementation

**Cons:**
1. Allows for phase change only along interface
2. Cannot maintain saturation temperature

#### Schrage Model (1953)

Kartuzova & Kassemi (2011), Magnini *et al.* (2013) …

**Pros:**
1. Successfully used for evaporating & condensing films

**Cons:**
1. Requires use of empirical coefficient $\gamma$
2. Allows for phase change only along interface

#### Lee Model (1980)


**Pros:**
1. Ease of implementation
2. Successfully used for condensation processes

**Cons:**
1. Not applicable for subcooled boiling
2. Requires use of empirical coefficient $r_i$
Computational Domain, Governing Equations and Boundary Conditions

**Computational Domain**
- Modeled axi-symmetric region
- Cooling wall
- Uniform mesh near wall
- Gradually finer mesh near wall
- Uniform mesh

**Governing Equations**

- Continuity Equations
  - Liquid Phase:
    \[ \frac{\partial}{\partial t} \left( \alpha_f \rho_f \right) + \nabla \times \left( \alpha_f \rho_f \bar{u}_f \right) = S_f \]
  - Vapor Phase:
    \[ \frac{\partial}{\partial t} \left( \alpha_g \rho_g \right) + \nabla \times \left( \alpha_g \rho_g \bar{u}_g \right) = S_g \]
    \[ S_g = -S_f = r_i g \left( \frac{T - T_{sat}}{T_{sat}} \right) \]

- Momentum Equation
  \[ \frac{\partial}{\partial t} (\rho \bar{u}) + \nabla \times (\rho \bar{u} \bar{u}) = -\nabla P + \nabla \left[ \mu (\nabla \bar{u} + \nabla \bar{u}^T) \right] + \rho g \bar{g} + \bar{F} \]

- Energy Equation
  \[ \frac{\partial}{\partial t} (\rho E) + \nabla \times (\bar{u} (\rho E + P)) = \nabla \times (k_{eff} \nabla T) + Q \]
  \[ Q = h_{fg} S_f \]

**Boundary Condition**
- Axisymmetric centerline
- \( k-\omega \) SST turbulence model
- Inlet uniform velocity from experimental data
- Wall heat flux from experimental data
Void Fraction Results: Climbing Film Regime

Climbing film

G = 106.5 kg/m²s
x_{e,in} = 1.16

G = 116.7 kg/m²s
q_{wall,avg} = -3.10 W/cm²
x_{e,in} = 1.16
Predictions of Average Heat Transfer Coefficient

$\dot{h}_{avg}$ [W/m$^2$K] vs $G$ [kg/m$^2$s]

- Experimental
- Computational

FC-72
Future Computational Modeling
- Fluent is able to accurately replicate experimental results, but "tuning" necessary for phase change model means it is not a reliable predictive tool
- Difficult to work with Fluent because solver code is proprietary
- Fluent very robust and can tackle wide range of problems, making it slower than a dedicated research code
- Fluent does not utilize cutting edge multi-phase computational techniques
Current Work

- Working with Prof. Carlo Scalo’s group at Purdue University to develop a 2-D code using best available computational techniques
- Performing comparisons with Fluent to quantify in-house solver performance

Future Work

- Use proposed 2-D code to run select cases which can be represented reasonably well by 2-D domains (e.g., axi-symmetric flow condensation, slug flow)
- Scale 2-D code up to 3-D, highly parallelized solver, which will include turbulence effects, to be run on Purdue supercomputing cluster
- Begin comparing data for transient cases with 3-D geometry to prior experimental studies
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