Abstract

PEEPLES, JOHANNA LOUISE. Design and Optimization of Thermosyphon Batch Targets for Production of $^{18}$F. (Under the direction of Joseph Michael Doster.)

$^{18}$F is a short-lived radioisotope commonly used in Positron Emission Tomography (PET). This radionuclide is typically produced through the $^{18}O(p,n)^{18}F$ reaction by proton bombardment of $^{18}$O-enriched water. Thermosyphon batch targets have been proposed as a means to increase $^{18}$F production due to their enhanced heat rejection capabilities. These boiling targets have been operated with up to 3.2 kW of beam power with manageable $^{18}$O enriched water volumes. The purpose of this research project has been to develop computational methods which can be used to design new targets with enhanced production capabilities. The computational methods developed in this work were used to design a low power thermosyphon production target for the Duke Medical Center cyclotron. This design was modeled to be range thick, and operate within the desired margins for beam powers in excess of 1 kW, the operating limit of the Duke cyclotron. A sensitivity analysis of the computational methods was performed which indicated the model is most sensitive to the boiling and condensing heat transfer coefficients. Even with a high uncertainty in these coefficients, the target should still operate well within the desired margins.
DESIGN AND OPTIMIZATION OF THERMOSYPHON BATCH TARGETS FOR PRODUCTION OF $^{18}$F

by

JOHANNA LOUISE PEEPLES

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APPROVED BY:

_________________________        _________________________
Dr. T. Gerig                 Dr. M. Bourham

_________________________        _________________________
Dr. J.M. Doster                 Dr. B. Wieland
Chair of Advisory Committee

Approved
For Cody
**Biography**

Johanna Louise Peeples was born in Durham, North Carolina in May of 1985 to Jane Cornelius and Wayne Cornelius. She was educated in the public school system and graduated from William G. Enloe High School in Raleigh, NC in 2002. Following her early graduation from high school, she began her college career at North Carolina State University. She graduated in May of 2005, with a Bachelor of Science degree in Nuclear Engineering, as a university valedictorian. She continued on at North Carolina State University to obtain her Master of Science degree in Nuclear Engineering. While pursuing her degree, she wed Cody Ryan Peeples in June of 2006. After graduation in December 2006, she is continuing her studies in the Nuclear Engineering Department at North Carolina State University, with the intention to pursue a Ph.D.
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List of Symbols and Abbreviations

\( A \) – area (\( \text{ft}^2 \))
\( A_x \) – cross-sectional area (\( \text{ft}^2 \))
\( A_\omega \) – surface area associated with \( \omega \) (\( \text{ft}^2 \))
\( \alpha \) – void fraction
\( \alpha_g \) – vapor void fraction
\( \alpha_l \) – liquid fraction
\( Bo \) – Bond number
\( \text{boil} \) (subscript) – refers to boiling
\( \text{boil1} \) (subscript) – refers to boiling in radial direction
\( \text{boil2} \) (subscript) – refers to boiling on back plate
\( C_o \) – Wallis Equation coefficient
\( C_p \) – specific heat (Btu/lbm-F)
\( CD \) (subscript) – refers to condensing
\( CD1 \) (subscript) – refers to condensing in radial direction
\( CD2 \) (subscript) – refers to condensing on back plate
\( d \) – jet exit diameter (ft)
\( D \) – diameter (ft)
\( D_e \) – equivalent diameter (ft)
\( D/d \) – ratio of the diameter of the impingement surface to the jet exit diameter
\( -dE/dx \) – stopping power (MeV/cm)
\( E \) – proton energy (MeV)
\( f \) – friction factor
\( \text{FDG} \) – fluorodeoxyglucose
\( g \) – acceleration due to gravity (32.2 \( \cdot 3600^2 \) ft/hr^2)
\( g_c \) – conversion factor (32.2 \( \cdot 3600^2 \) ft-lbm/lbF-hr^2)
$G$ – mass flux (lbm/hr-ft$^2$)
$G_p$ – mass flux in piping (lbm/hr-ft$^2$)
$G_r$ – mass flux in target channels (lbm/hr-ft$^2$)
$G_{\omega}$ – mass flux at location $\omega$ (lbm/hr-ft$^2$)
$h_{\text{boil}}$ – pool boiling heat transfer coefficient (Btu/hr-ft$^2$-F)
$h_c$ – heat transfer coefficient (Btu/hr-ft$^2$-F)
$h_{\text{cool}}$ – radial coolant heat transfer coefficient (Btu/hr-ft$^2$-F)
$h_{fg}$ – enthalpy change by evaporation (Btu/lbm)
$h'_{fg}$ – modified enthalpy change by evaporation (Btu/lbm)
$h_{\text{jet}}$ – submerged jet impingement heat transfer coefficient (Btu/hr-ft$^2$-F)
$h_{\text{sub}}$ – heat transfer coefficient associated with sub (Btu/hr-ft$^2$-F)
$H$ – total height (cm)
$H_b$ – boiling height (cm)
$H_{cd}$ – condensing height (cm)
$I$ – beam current (µA)
$k$ – thermal conductivity (Btu/hr-ft-F)
$K_{\omega}$ – forms loss coefficient associated with $\omega$
$L$ – length (m)
$m$ – mass flow rate (lbm/hr)
$\mu$ – dynamic viscosity (lbm/ft-hr)
$Nu$ – Nusselt number
$\nu_p$ – kinematic viscosity (ft$^2$/hr)
$P_{\text{exit}}$ – exit pressure (psi)
$P_{\text{in}}$ – inlet pressure (psi)
$P_{\omega}$ – pressure at location $\omega$ (psi)
$\Delta P$ – pressure difference between manifolds (psi)
PET – positron emission tomography
Pr – Prandtl number

$P_z$ – pressure (psi)

$q''(x)$ – volumetric heat generation rate (Btu/hr-ft$^3$)

$q''_{avg}$ – average heat flux (Btu/hr-ft$^2$)

$q''_{crit}$ – critical heat flux (Btu/hr-ft$^2$)

$q''_{max_z}$ – Zuber-Kutateladze critical heat flux (Btu/hr-ft$^2$)

$q''_{peak}$ – peak heat flux (Btu/hr-ft$^2$)

$\dot{Q}$ – total heat input (W)

$\dot{Q}_{b,j}$ – heat transfer rate from boiling region into jet (W)

$\dot{Q}_{b,r}$ – heat transfer rate from boiling in radial direction (W)

$\dot{Q}_{CD,j}$ – heat transfer rate from condensing region into jet (W)

$\dot{Q}_{CD,r}$ – heat transfer rate from condensing region in radial direction (W)

$\dot{Q}_r$ – total heat transfer rate (W)

$\dot{Q}_{\omega}$ – heat transfer rate associated with $\omega$ (W)

$Ra$ – Rayleigh number

$Re$ – Reynolds number

$R_{pta}$ – peak-to-average ratio

$\rho$ – density (lbm/ft$^3$)

$\rho_f$ – fluid density (lbm/ft$^3$)

$\rho_{fg}$ – density change by evaporation (lbm/ft$^3$)

$\rho_g$ – vapor density (lbm/ft$^3$)

$S$ – jet exit-to-impingement distance (ft)

$S/d$ – ratio of the exit-to-impingement distance to the jet exit diameter

$\sigma$ – surface tension (lbF/ft)

$T$ – temperature (°F)

$T_{cool}$ – coolant temperature (°F)
\( T_f \) – film temperature (°F)

\( T_{\text{max}} \) – maximum temperature in window foil (°F)

\( T_{\text{sat}} \) – saturation temperature (°F)

\( T_{\text{wall}} \) – wall temperature (°F)

\( T_{\infty} \) – bulk fluid temperature (°F)

\( T_{\omega} \) – temperature at location \( \omega \) (°F)

TS – thermosyphon

\( v_g \) – vapor velocity (ft/hr)

\( v_f \) – fluid velocity (ft/hr)

\( v_{\omega} \) – velocity at location \( \omega \) (ft/hr)

\( W \) – width (ft)

WMC – Wisconsin Medical Cyclotron

\( x_{\text{max}} \) – axial location of maximum temperature in window foil (ft)

\( \zeta \) – expression which takes values \( \in [0,1] \)
Chapter 1

Introduction

1.1 Background

Positron emission tomography (PET) is a medical imaging technique that can provide a detailed map of molecular activity and biology in patients. PET has applications in oncology, neurology, cardiology, psychiatry, and pharmacology. In the field of oncology, fluorodeoxyglucose (FDG), a sugar, is used as a radiotracer. FDG is tagged with the positron emitting radioisotope Flourine-18 (\(^{18}\text{F}\)), which can be detected with PET scanning. When injected into the human body, FDG is absorbed and metabolized in cells which use glucose, releasing the \(^{18}\text{F}\) which remains trapped in the cell. Higher uptake occurs in certain tissues, including the brain and liver and cancerous cells. As a result, these high activity cells appear as bright spots on a PET image (Valk, 2003).

The physics of PET scanning is based on the positron emission of certain radionuclides. During this nuclear decay, a positron, or positively charged electron, is emitted. The positron travels only a short distance, 0.6 mm on average in human tissue for positrons emitted from \(^{18}\text{F}\), before colliding with an electron. This collision results in an annihilation of the electron-positron pair and leads to the emission of two annihilation photons. These photons each have 511 keV of energy, corresponding to the rest mass of the annihilated particles, and travel along a straight line in opposite directions. A PET scanner detects the two photons in coincidence and creates a map from the intersecting decay lines (Valk, 2003).
$^{18}$F is a short-lived radioisotope, which decays by positron emission with a half-life of 109.77 minutes. This radionuclide is commonly produced through the $^{18}O(p,n)^{18}F$ reaction. A cyclotron is used to accelerate protons above the 2.4 MeV reaction threshold. The protons are focused into a beam and directed into a small volume of $^{18}$O-enriched water. The $^{18}$F produced is then chemically synthesized into FDG. Due to the short half-life, the production must occur in a cyclotron with a short delivery-time to the PET scanner. The $^{18}$F activity can decay significantly over the course of a day, so PET facilities require either a cyclotron on-site or a nearby FDG distribution center (Nuclides, 1996).

Thermosyphon batch targets are one type of target used for $^{18}$F production. A proton beam enters the water and deposits heat as the protons slow down. The target water is permitted to boil during this process, such that under steady-state operation, a condensing region forms above the boiling pool. The volume associated with the boiling region and with the condensing region is governed by the heat input and available cooling systems (Wieland, 2002).

Adequate cooling of the target is necessary to avoid over-pressurization of the target chamber or excessive voiding. Excessive voiding could allow the proton beam to penetrate to the back of the target, which would greatly reduce the yield of $^{18}$F. Beam penetration to the back of the target could also damage the walls of the chamber, and/or adversely affect the chemistry of the FDG produced.

**1.2 Purpose**

The purpose of this project is to develop computational methods which can be used to design new targets with enhanced production capabilities. Prior to this work, design of these targets was purely empirical, which required a significant amount of trial and error, long lead
times and no guarantee of an optimal design. Due to the high cost of enriched water ($50-100 per gram), it is desirable for new targets to produce the maximum activity of viable $^{18}$F with the minimum liquid volume. The production rate of $^{18}$F is directly proportional to the proton beam current. As a result, the maximum $^{18}$F activity will be produced using the maximum feasible beam current. The amount of heat deposited into the liquid volume also depends directly on the beam current. The heat input corresponds to the beam power, the product of the beam current and the proton energy.

If heat input exceeds the heat removal capability of the target, excessive voiding can occur in the target water. Protons of a given energy have a characteristic range in water which is inversely related to the water density. Since water density decreases with void fraction, the operating void fraction dictates the necessary target depth to prevent the beam from penetrating to the target back. If the proton beam penetrates the water volume and deposits heat in the back wall of the target, the $^{18}$F yield will be reduced due to less proton interactions in the water. Interactions between the beam and the target wall can also release ions into the water, which react with the ionic $^{18}$F to further reduce the yield. Beam penetration can not only reduce yields in the current batch, but can result in long-term or permanent contamination of the production target.

Adequate cooling of the target can prevent excessive voiding and prevent beam penetration. The optimal cooling configuration maximizes heat removal while minimizing the target water volume. This allows for the highest heat input, or highest acceptable beam current, and therefore produces the most $^{18}$F. Minimizing the volume of enriched water needed to provide these conditions reduces the production costs. The optimal target chamber dimensions and cooling configuration can be determined for a given cyclotron, with given
beam characteristics. The design methods developed during this study were applied to
design an optimized target for the Cyclotron Corp. CS-30 cyclotron at the Duke University
Medical Center in Durham, NC. The same methods will be applied to design a 2.4 kW target
for the Wisconsin Medical Cyclotron in Milwaukee, WI.

1.3 Related Work

Enriched $^{18}$O water targets have been under development for more than 20 years.
Conventional production targets are boiling liquid targets, which operate in batch mode.
Many of these targets are used on small cyclotrons which can produce beam powers in the
range of 300 to 1000 W. More than half of the installed base of PET cyclotrons in the US
can produce higher beam powers, but losses associated with boiling have limited all
conventional targets to operation below 1 kW. Target construction materials and associated
water contamination has been shown to affect $^{18}$F yields (Alvord, 2005).

Target yields at 79% theoretical maximum have been observed for a silver body, gold
back target with small water volume and beam power up to 440 W (Roberts, 1995). Good
yields have also been observed for a double foil, low pressure production target with silver
body and Havar window foils at 340 W beam power (Berridge, 1999). A spherical niobium
target with external water cooling has been operated at 650 W to produce 95% theoretical
yield (Zeisler, 2000). Increased production yields have also been demonstrated using single
foil targets constructed of niobium and titanium. These large volume, low pressure targets
operate at beam power up to 850 W (Berridge, 2002). In recent years, tantalum has been
used as a target material, due to its good oxidation resistance. Production yields have been
increased using a compact tantalum target with maximum beam power approaching 1 kW
(Alvord, 2005).
Bruce Technologies, Inc. has been investigating the use of thermosyphon batch boiling targets for $^{18}$F production for the past six years (Wieland, 2006). The targets are initially filled with water, and pressure is applied to the bottom of the target to produce a self-regulating condensing surface. Early thermosyphon targets with 1 mL water volume operated at 440 W beam power. Experiments have been performed at higher beam powers, and enhancements have been made in successive designs to increase yield. Experiments have indicated that thermosyphon targets are capable of operation with beam power in excess of 1 kW (Roberts, 2002). A high power thermosyphon target is currently under design for the Wisconsin Medical Cyclotron which will operate at beam power around 2.4 kW.

Alternate target designs have been proposed to increase production of $^{18}$F. Recirculating targets which remove heat through an external heat exchanger have been demonstrated to reject heat inputs above 2.7 kW at atmospheric pressure (Clark, 2004). Recirculating targets require greater water volumes than boiling targets but are capable of operating at much higher beam powers (Newnam, 2006).
Chapter 2

Design Constraints

2.1 Materials

The major components of a thermosyphon target include the target chamber, target body or housing and the target window.

Figure 1: TS-5 Assembly Drawing (Mark Humphrey 2005)
Table 1: TS-5 Target Parts List

<table>
<thead>
<tr>
<th>Item</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Shadow Grid (Body)</td>
</tr>
<tr>
<td>2</td>
<td>Front Flange (Body)</td>
</tr>
<tr>
<td>3</td>
<td>Target Chamber</td>
</tr>
<tr>
<td>4</td>
<td>Target Chamber Back</td>
</tr>
<tr>
<td>5</td>
<td>Intermediate Flange (Body)</td>
</tr>
<tr>
<td>6</td>
<td>Target Flange (Body)</td>
</tr>
<tr>
<td>7</td>
<td>Back Flange (Body)</td>
</tr>
</tbody>
</table>

The target window is a thin metal foil through which the proton beam enters the target chamber. The target window should be thin enough to allow protons to enter with minimum attenuation, generally 12 to 50 μm, but must be thick enough to withstand the high target operating pressure. Use of a light material is also desirable to minimize beam attenuation. The window material should have high mechanical strength, high resistance to radiation damage, and a high melting point. High thermal conductivity is preferred to facilitate cooling of the foil. Because the foil is in contact with the target water, a material should be selected to minimize chemical contamination of the water and synthesized FDG. Common window materials include beryllium copper, titanium, tantalum, and most frequently a proprietary alloy called Havar.

The target chamber internals are also in contact with the target water. Chemical compatibility, i.e. low potential for contamination, is a key concern. High thermal conductivity, for effective cooling, is also highly preferred. As with the window, high melting point, high material strength, and good corrosion resistance are important. Beam interactions with the metal foil and the metal target internals can result in formation of metal ions. The ions can interact with the $^{18}$F, which is also produced in an ionic form. Contamination from the target materials pollutes the target water and can significantly reduce the yield of $^{18}$F.
The target housing supports the target window and internals and allows it to be properly positioned on the cyclotron beam line. The target housing should be durable, have high strength, and high resistance to radiation damage and activation. During long production runs with high beam currents, activation can be significant.

In the past, silver has been used for the target chamber due to its high thermal conductivity and above average chemical compatibility. Aluminum has also been used in target bodies due to its high thermal conductivity, ease of machining, and low cost. Poor chemical compatibility, however, required the use of a protective coating or film to separate the target water from the aluminum. Tantalum is an attractive interface material due to its excellent chemical compatibility, but suffers from a relatively low thermal conductivity. Tantalum inserts, press fit into an aluminum target body, were examined, but early designs exhibited high thermal resistance due to large conduction distances and high contact resistance at the aluminum/tantalum interface. Experiments by researchers at Duke University are examining the feasibility of chemical vapor deposition coating of aluminum with tantalum. Successful sputtering of tantalum onto aluminum could yield a surface with the chemical inertness of tantalum and the high thermal conductivity of aluminum. While the feasibility of this method is being determined, target designs are being developed with internals constructed of pure tantalum. Tantalum is expensive, difficult to machine and has much lower thermal conductivity, but the chemical compatibility must take precedence to ensure a high $^{18}$F yield. Aluminum can be used for the target housing due to its low cost and ease of machining. Since the target cooling systems can be located completely within the tantalum internals, good thermal contact between the internals and the housing is not necessary.
2.2 Particle Energy

The Duke Medical Center cyclotron accelerates protons to a maximum of 27 MeV and focuses them into a narrow beam. The protons in the beam lose a small amount of energy when they pass through the window into the target chamber. The protons then lose additional kinetic energy as they travel through the water due to charged particle interactions and bremsstrahlung. Bremsstrahlung losses are radiation losses that occur as a charged particle slows down in a material.

The incremental energy loss of the proton per unit of distance traveled in the media \(-\frac{dE}{dx}\) is defined to be the stopping power. The stopping power changes with the energy of the proton as it travels farther into the water, and can be shown to increase to a peak near the end of the particle path and then decrease suddenly to zero (Faw, 1999).

![Figure 2: Variation of Stopping Power and Energy along Charged Particle Path (Faw, 1999)](image-url)
Figure 3: Collisional Stopping Power for Protons in Air, Water, Aluminum, and Iron (Faw, 1999)
The majority of the heat is deposited in the region where the stopping power is the greatest. With the deposition of heat, the water density decreases and vapor voids are formed. Higher incident proton energies result in lower stopping power, and therefore deeper penetration into the water volume. The range of the protons is the average distance that they will travel in the water before being stopped (Faw, 1999).

To prevent the proton beam from striking the back wall of the target, the target must be range thick. In other words, the target depth should exceed the range of the protons in the saturated mixture. The range of protons with incident energy between 8 and 30 MeV in water at 300 psi saturation conditions and various void fractions was determined using The Stopping and Range of Ions in Matter program (SRIM). The desired operating pressure of the Duke target, 300 psi, was chosen to limit stress on the beam window and accommodate the pressure rating of the valves in the target system.

Figure 4: Range Thickness for Protons in 300psi Saturated Water (Matthew Stokely 2006)
The depth of the target chamber dictates the maximum amount of voiding that can be tolerated. A depth of 15 mm was selected for the Duke target by the Cyclotron Director, Dr. Gerald Bida. During operation of the Duke target, protons will be accelerated to 22 MeV, rather than the 27 MeV maximum. From Figure 5, this implies the target should be designed to ensure that the average void fraction would never exceed 60%.

2.3 Heat Input

The heat input $Q$ to the target is a function of the beam current and the energy of the incident protons. Beam current $I$ is quantified in micro-amps ($\mu$A) and proton energy $E$ is quantified in mega-electron volts (MeV).
\[ \dot{Q}(\text{Watts}) = I(\mu A) \cdot \left( \frac{10^{-6} \text{ C/s}}{1 \mu A} \right) \cdot \left( \frac{1}{1.6 \cdot 10^{-19} \text{ C/}p^+} \right) \cdot E \left( \frac{\text{MeV}}{p^+} \right) \cdot \left( \frac{1.6 \cdot 10^{-13} \text{ J}}{1 \text{ MeV}} \right) \cdot \left( \frac{1 \text{ W}}{1 \text{ J/s}} \right) \]

The beam power in Watts, which is equivalent to the heat input, can be determined directly by taking the product of the beam current in \( \mu A \) and the incident proton energy in MeV.

The intended operation conditions for the Duke target are 22 MeV protons at a beam current of 25 \( \mu A \). This corresponds to a proposed heat input of 550 Watts. For conservatism, the target was designed to perform within the desired specifications for a heat input on the order of 750 Watts.

### 2.4 Thermal Limit

Operation of a thermosyphon target is illustrated in Figure 6. A thermosyphon target is designed to allow considerable boiling in the target chamber to take advantage of the high rate of heat transfer. Heat deposition in the water causes boiling, and results in the formation of voids or bubbles. Buoyancy forces cause these bubbles to rise and enter the condensing region of the target chamber. Steam condenses on the chamber walls and flows back down the walls into the boiling pool. Under steady-state operation, there will be a pool of boiling water with some height \( H_b \) and above it will be a completely voided condensing region with some height \( H_{cd} \). These regions compose the total height of the target chamber \( H = H_b + H_{cd} \). Thermosyphon targets are self-regulating, in that the water level, or boiling height, will change according to the heat input. The water level will settle at a location where the amount of heat being removed via cooling equals the heat input.
Protons accelerated in the cyclotron are focused into a narrow beam. The optics of this beam vary for each cyclotron and must be characterized for target design. The proton beam passes through a circular window and enters the target water volume. The Duke target requires a 10 mm diameter beam window. This window is positioned near the bottom of the target to ensure that the beam enters the boiling region of the water volume rather than the condensing region.

If the boiling water level drops below the height of the beam window, there will be insufficient water density to maintain range thickness and some of the beam can penetrate to the target back. The thermal limit of a target is then defined to be the minimum heat input that will result in either (1) the average void fraction in the boiling region exceeding the value necessary for the target to remain range thick or (2) the boiling height falling below the height of the beam window. For the Duke target these values are 60% and 10 mm, respectively. It should be noted, that operation of the target is still possible above the defined thermal limit, though $^{18}$F yields will suffer and ultimately target damage can occur.
Chapter 3

Optimization of Radial Coolant Channels

3.1 Description

Cooling for the target is provided by two independent systems as illustrated in Figure 7. Radial coolant channels are located around the target chamber and run parallel to the target chamber and the direction of the beam. Additional cooling is provided by a jet of water which impinges on the back of the target. Heat deposited in the target causes the water temperature to rise and heats the walls of the target chamber. This heat conducts through the walls of the chamber and is then removed by either the radial coolant channels or the jet system. Many commercial targets rely on cooling through a single jet on the back of the target. The use of radial cooling can result in increased heat removal capability for the same water volume.
Figure 7: Axial View of Target Cooling Systems (Mark Humphrey and Johanna Peeples 2006)
Figure 8: Radial View of Target Cooling Systems (Mark Humphrey and Johanna Peeples 2006)
3.2 Pressure Losses

The radial coolant flow is extracted from a manifold which provides cooling water to the cyclotron. The Duke Cyclotron facility has a large volume of 60°F water with a pressure difference of about 71 psi between the inlet and outlet manifolds. Pressure losses in the plastic piping connecting these two manifolds have been measured experimentally.

![Diagram showing pressure difference across manifolds for Duke Cyclotron Center](image)

Other pressure losses occur within the radial channels themselves. The flow path associated with these channels has been greatly simplified from earlier target designs in order to minimize these losses. The pressure losses in the radial channels can be estimated using characteristic forms losses as indicated in Equations 1 through 7. With this information and the known pressure difference, the mass flow rate through the radial channels can be estimated. This mass flow rate, along with the known coolant temperature, will determine the heat removal capability of the radial channels.
Figure 10: Mass Flow through Radial Coolant Channels

The pressure drop through each segment of the path can be written as

\[ P_{in} = P_1 + \left( \frac{fL_1}{D} + K_1 \right) \frac{G_p^2}{2 \rho g_c} \]  
\[ (1) \]

\[ P_1 = P_{man1} + K_{ex} \frac{G_p^2}{2 \rho g_c} \]  
\[ (2) \]

\[ P_{man1} = P_2 + K_{in} \frac{G_T^2}{2 \rho g_c} \]  
\[ (3) \]

\[ P_2 = P_3 + \left( \frac{fL}{D} \right) \frac{G_T^2}{2 \rho g_c} \]  
\[ (4) \]

\[ P_3 = P_{man2} + K_{ex} \frac{G_T^2}{2 \rho g_c} \]  
\[ (5) \]

\[ P_{man2} = P_4 + K_{in} \frac{G_p^2}{2 \rho g_c} \]  
\[ (6) \]

\[ P_4 = P_{ex} + \left( \frac{fL_2}{D} + K_2 \right) \frac{G_p^2}{2 \rho g_c} \]  
\[ (7) \]
In Equations 1 through 7, the subscript $p$ refers to the flows in the smooth plastic piping that connects the water manifolds to the target. The subscript $T$ refers to the flows within the target. The sum of Equations 1 through 7 is

$$P_{in} - P_{exit} = \left(\frac{fL}{D}\right)_p + K_p + K_{ex} + K_{in} \left\{ \frac{G_p^2}{2\rho g_c} + \left(\frac{fL}{D}\right)_T \right\} \left\{ \frac{G_T^2}{2\rho g_c} \right\}$$

(8)

where $L_p = L_1 + L_2$ and $K_p = K_1 + K_2$.

The friction factor for smooth piping, as a function of the Reynolds number is

$$f = \begin{cases} 
64/\text{Re} & \text{Re} \leq 2300 \\
0.316\text{Re}^{-0.25} & 2300 < \text{Re} < 20000 \\
0.184\text{Re}^{-0.2} & \text{Re} > 20000
\end{cases}$$

$$f = a \text{Re}^b$$

(Incropera, 2007)

Equation 8 can be rewritten as

$$\Delta P = \left(\frac{a \text{Re}^b L_p}{D} \right)_p + K_p + K_{ex} + K_{in} \left\{ \frac{G_p^2}{2\rho g_c} + \left(\frac{a \text{Re}^b L_T}{D}\right)_T \right\} \left\{ \frac{G_T^2}{2\rho g_c} \right\}$$

(9)

Mass flux through the piping that connects the water manifolds to the target and mass flux through the target are related due to conservation of mass. This relation can be written as

$$G_p A_p = G_T A_{T-total} = G_T n_T A_T = \dot{m}$$

(10)

An experiment was conducted to estimate the forms loss coefficient in the cyclotron manifold piping. The piping between the manifolds was joined with no target present, and the mass flow rate through the piping was measured using a flow meter. The forms loss for the cyclotron manifold piping was then estimated using the following equation.

$$\Delta P = \left(\frac{a \text{Re}^b L_p}{D_p} + K_p \right) \frac{G_p^2}{2\rho g_c}$$

(11)
Equation 11 can be solved for the loss coefficient in the piping to yield

\[
K_p = \Delta P \left( \frac{2 \rho \gamma}{G_p^2} \right) - \left\{ a \left( \frac{G_p D_p}{\mu} \right)^b \frac{L_p}{D_p} \right\} 
\]

(12)

Volumetric flow rate was measured to be

\[
\frac{\dot{m}}{\rho} = 5.5 \frac{gal}{min} \approx 35.5 \frac{ft^3}{hr}
\]

This corresponds to mass flux

\[
G_p = \frac{\dot{m}}{A_x} = \frac{\dot{m} \left( \frac{\rho}{A} \right)}{\rho} = \frac{\left( \frac{35.5}{hr} \right) \left( \frac{62.3 \text{ lbm}}{ft^3} \right)}{\left( \pi (0.125/12)^2 ft^2 \right)} = 6.49 \cdot 10^6 \frac{\text{lbm}}{hr - ft^2}
\]

The experimentally determined forms loss coefficient for the piping is

\[
K_p = \left( 71.144 \frac{\text{lbF}}{ft^2} \right) \left( 2.623 \frac{\text{lbm}}{ft^3} \right) \left( 32.2 \cdot 3600 \frac{ft - \text{lbm}}{hr - \text{lbF}} \right) \left( 6.49 \cdot 10^6 \frac{\text{lbm}}{hr - ft^2} \right)^{-0.2} 
\]

\[
-0.184 \left( \frac{6.49 \cdot 10^6 \frac{\text{lbm}}{hr - ft^2}}{2.731 \frac{\text{lbm}}{ft - hr}} \right)^{8 ft \over 0.25/12 ft} 
\]

\[
K_p = 4.5
\]

The forms loss coefficient associated with a sudden contraction is

\[
K_{in} = 0.5 \left( 1 - \frac{A_{\text{small}}}{A_{\text{large}}} \right)^{0.75} \quad \text{(Idelchik, 1994)}
\]

For a very severe restriction in the flow area, this loss coefficient reduces to a maximum with

\[
K_{in} = 0.5
\]

The forms loss coefficient associated with a sudden expansion is
\[ K_{ex} = \left(1 - \frac{A_{small}}{A_{large}}\right)^2 \]  
(Idelchik, 1994)

For a very large expansion in the flow area, this coefficient approaches a maximum with

\[ K_{ex} = 1.0 \]

The mass flux \( G \) and Reynolds (Re) number associated with each section of piping can be written in terms of the mass flow rate. The Reynolds number as a function of mass flow rate is

\[ \text{Re}_p = \frac{G_p D_p}{\mu_p} = \frac{\dot{m} D_p}{\mu_p A_p} \]  \hspace{1cm} (13)

and

\[ \text{Re}_T = \frac{G_T D_T}{\mu_T} = \frac{\dot{m} D_T}{\mu_T n_T A_T} \]  \hspace{1cm} (14)

Equation 9 can be rewritten as

\[ \Delta P = \left\{ \frac{L_p}{D_p} a_p \left( \frac{D_p}{\mu A_p} \right)^{b_p} m^{b_p} + K_p + K_{ex} + K_{in} \right\} \frac{m^2}{2 \rho g \epsilon A_p^2} + \left\{ K_{in} + K_{ex} + \frac{L_T}{D_T} a_T \left( \frac{D_T}{\mu A_T} \right)^{b_T} \right\} \frac{m^2}{2 \rho g \epsilon A_T^2} \]  \hspace{1cm} (15)

Equation 15 can be solved implicitly for the mass flow rate.

### 3.3 Heat Transfer Coefficient

The heat removal capability of the radial channels can be estimated using a heat transfer coefficient \( h_{cool} \) and the known coolant fluid temperature. The Dittus-Boelter equation is a well known correlation for heat transfer coefficients for flow in circular pipes. The heat transfer coefficient is a function of the Reynolds number, and is higher for higher
mass flow rate. For situations where the fluid inside the pipe is being heated, i.e. it is cooling the walls of the pipe, the Nusselt number is

\[ Nu = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \]  

(Incropera, 2007)

Using the definition of the Nusselt number, this equation becomes

\[
\frac{h_{\text{cool}} D_e}{k} = 0.023 \left( \frac{\dot{m} D_e}{A_s \mu} \right)^{0.8} \left( \frac{C_p \mu}{k} \right)^{0.4}
\]

such that the heat transfer coefficient for fluid cooling the walls of a radial cooling channel is

\[
h_{\text{cool}} = 0.023 \frac{k}{D_e} \left( \frac{\dot{m} D_e}{A_s \mu} \right)^{0.8} \left( \frac{C_p \mu}{k} \right)^{0.4}
\]

### 3.4 Conduction Distance

An optimization study was performed to determine the ideal coolant channel configuration. Previous thermosyphon targets incorporated 16 radial coolant channels of 0.188” diameter. A study was performed where the 16 coolant channels were moved from their position in the old design inward towards the wall of the chamber, thereby reducing the conduction distance. The diameter of the coolant channels was scaled along with the conduction distance to maintain a channel to channel spacing of one channel radius. A representative 1/16 radial wedge was modeled in COMSOL Multiphysics as illustrated in Figure 11. The inner surface of the wedge was held at 444.59°F, the saturation temperature of water at 400 psi. The mass flow rate for each proposed configuration was calculated from Equation 15, and the corresponding heat transfer coefficient was applied to the coolant channel boundary. The other boundaries were isolated thermally.
The linear heat rate, the two-dimensional analog to the heat input, was determined for each of these configurations and is given in Figure 12.

![Figure 11: COMSOL Temperature Profiles for 1/16 Target Wedges](image)

![Figure 12: Linear Heat Rate vs. Coolant Channel Diameter](image)
It was shown that the best thermal performance, i.e. highest linear heat rate, was observed for small closely-spaced coolant channels. This result indicates that the low thermal conductivity of the tantalum is the highest resistance to the heat transfer. Reduction in flow rate, and therefore the heat transfer coefficient, associated with moving to smaller piping was more than compensated for by the accompanying reduction in the conduction distance. As a result, the conduction distance was determined based on the minimum distance that could be machined with acceptable tolerance. In order to achieve very small diameters, the coolant channels will be machined using the wire EDM technique. This method uses a thin electrode that follows a programmed path, and is able to cut conductive materials with high accuracy (XACT, 2006). A minimum distance quoted by XACT Wire EDM Corp is 0.01”.

3.5 Number and Spacing

Additional studies were performed to determine the ideal number and associated channel-to-channel spacing of the coolant channels. The addition of more channels, at closer channel-to-channel spacing, resulted in improved thermal performance (Figure 2). To optimize thermal performance, the minimum spacing that could be machined with appropriate tolerances, 0.01”, was selected. When evaluating potential target heights, the number of coolant channels was selected to be the maximum number that could be positioned with channel-to-channel spacing no smaller than 0.01”. 
Table 2: Linear Heat Rate for Radial Coolant Geometry

<table>
<thead>
<tr>
<th>Point</th>
<th>Conduction Distance (in)</th>
<th>Diameter (in)</th>
<th>Number</th>
<th>Linear Heat Rate (Relative)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3/16</td>
<td>1/16</td>
<td>26</td>
<td>1.00</td>
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<tr>
<td>2</td>
<td>3/16</td>
<td>2/16</td>
<td>14</td>
<td>0.93</td>
</tr>
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<td>10</td>
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</tr>
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<td>3/16</td>
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<td>10</td>
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</tr>
<tr>
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<td>3/16</td>
<td>4/16</td>
<td>10</td>
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</tr>
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<td>1/16</td>
<td>26</td>
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</tr>
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</tr>
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</tr>
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</tr>
<tr>
<td>20</td>
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<td>4/16</td>
<td>14</td>
<td>0.52</td>
</tr>
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</table>
Chapter 4

Optimization of the Jet Cooling System

4.1 Description

Additional cooling for the target is achieved by jetting 60°F water onto the target back. The jet flow is driven by the same cyclotron facility manifold pressure difference as the radial coolant channels, but runs on an independent flow line. This provides for the highest available mass flow rates for both cooling systems.

4.2 Heat Transfer Coefficient

Many correlations have been developed for heat transfer coefficients associated with cooling from submerged jets (Brdlik, 1965; Sitharamayya, 1969; Martin, 1977; and Womac, 1993). These correlations are generally based on the geometry of the jet and of the impingement surface. The geometry of this type of jet can be evaluated based on two dimensionless ratios: the ratio of the exit-to-impingement distance to the jet exit diameter \((S/d)\) and the ratio of the diameter of the impingement surface to the jet exit diameter \((D/d)\) (Womac, 1993).
A correlation has been proposed for submerged water jets with $S/d \leq 6.2$, $1.67 \leq D/d \leq 18.75$, and $2000 \leq \text{Re} \leq 20000$. The Nusselt number from this correlation is

$$Nt_D = 1.54 \text{Re}_D^{0.5} \text{Pr}^{0.33}$$

(Brdlik, 1965)

For submerged water jets with $2 \leq S/d \leq 12$, $5 \leq D/d \leq 15$, and $2000 \leq \text{Re}_d \leq 400,000$, another correlation for submerged jet impingement has been proposed. This correlation is preferred because it is valid for a much broader range of Reynolds numbers. This correlation provides that the Nusselt number is
An optimization study was performed using the Martin correlation to determine the geometric ratios which correspond to the highest jetting heat transfer coefficients, and therefore the best heat removal capability.

\begin{equation}
Nu = \frac{d}{D} \left[ \frac{2 - 4.4 \frac{d}{D}}{1 + 0.2 \left( \frac{S}{d} - 6 \right) \left( \frac{d}{D} \right)} \right] F(Re_d) Pr^{0.42} \quad \text{(Martin, 1977)}
\end{equation}

where

\begin{equation}
F(Re_d) = 2 Re_d^{0.5} \left[ 1 + \frac{Re_d^{0.55}}{200} \right]^{0.5}
\end{equation}

\begin{figure}
\centering
\includegraphics[width=\textwidth]{figure15.png}
\caption{Selected Results of the Jet Impingement Geometry Optimization Study}
\end{figure}
The geometry configuration which resulted in the maximum heat removal had $S/d = 2$ and $D/d = 5$. A given target height would specify the value of $D$, and the values of $S$ and $d$ could be selected using these optimized ratios.

4.3 Nozzle

The water delivery system at the Duke cyclotron involves a large length of plastic piping with 0.375” outer diameter and 0.25” inner diameter. This piping diameter must be reduced sufficiently to clear the entrance through the back flange of the target. Similarly, smaller piping diameter at the exit of the jet is desired for maximizing the jetting heat transfer coefficient. A nozzle could be used to reduce the interior piping diameter from 0.25” to the desired exit diameter. While this nozzle could appear at any location in the system, it is desirable to place it at the end of the piping just before the jet, to avoid additional pressure losses associated with flow in very small pipes.
4.4 Pressure Losses

The jet flow is driven by the same pressure header as the radial coolant channels, but on a separate flow line. As with the analysis of the radial channels, pressure losses in the system can be estimated for the jet flow geometry based on characteristic forms losses. There are pressure losses in the jet flow system associated with the large length of 0.25” piping, the decrease in piping diameter that occurs at the nozzle, the jet itself, a 90° turn, and the reduction in flow cross-sectional area associated with reentering the 0.25” piping. Using the known pressure difference between the inlet and outlet manifolds and taking into account these pressure losses, the mass flow rate through the jet flow system can be estimated. This
mass flow rate, along with the known coolant temperature, will determine the magnitude of
the jetting heat transfer coefficient, and therefore set the heat removal capability of the jet
cooling system.

\[
P_{in} = P_1 + \left( \frac{fL_1}{D} + K_1 \right) \frac{G_p^2}{2 \rho g_c}
\]  

(1)

\[
P_1 = P_2 + K_n \frac{G_2^2}{2 \rho g_c}
\]  

(2)

\[
P_2 = P_3 + K_{baffle} \frac{G_2^2}{2 \rho g_c}
\]  

(3)

\[
P_3 = P_4 + K_{90} \frac{G_3^2}{2 \rho g_c}
\]  

(4)

\[
P_4 = P_{exit} + \left( \frac{fL_2}{D} + K_2 \right) \frac{G_p^2}{2 \rho g_c} + K_{in} \frac{G_p^2}{2 \rho g_c}
\]  

(5)

The sum of Equations 1 through 5 is

\[
P_{in} - P_{exit} = \left( \frac{fL}{D} \right)_p + K_p + K_{in} \frac{G_p^2}{2 \rho g_c} + \left( K_n + K_{baffle} \right) \frac{G_2^2}{2 \rho g_c} + \left( K_{90} \right) \frac{G_3^2}{2 \rho g_c}
\]  

(6)

where
\[ L_p = L_1 + L_2 \text{ and } K_p = K_1 + K_2 \]

The forms loss coefficient associated with the nozzle can be determined using the Bernoulli equation. The pressure balance is

\[ P_1 + \frac{\rho v_1^2}{2} = P_2 + \frac{\rho v_2^2}{2} \quad (7) \]

The pressure at the nozzle entrance is given by

\[ P_1 = P_2 + \frac{P}{2} \left( v_2^2 - v_1^2 \right) = P_2 + K_n \frac{\rho v_2^2}{2} \quad (8) \]

Conservation of mass requires that the mass flow rate in each segment of the pipe be equal.

The two mass flow rates are

\[ \rho v_1 A_1 = \rho v_2 A_2 \quad (9) \]

The velocity in the first section of pipe is

\[ v_1 = v_2 \left( \frac{A_2}{A_1} \right) \quad (10) \]

When Equation 10 is substituted into Equation 8, the pressure balance becomes

\[ P_1 = P_2 + \left( 1 - \left( \frac{A_2}{A_1} \right)^2 \right) \frac{\rho v_2^2}{2} = \left( 1 - \left( \frac{\pi D_2^2 / 4}{\pi D_1^2 / 4} \right)^2 \right) \frac{\rho v_2^2}{2} = \left( 1 - \left( \frac{D_2}{D_1} \right)^2 \right) \frac{\rho v_2^2}{2} \quad (11) \]

From this equation, the forms loss coefficient associated with the nozzle is

\[ K_n = 1 - \left( \frac{D_2}{D_1} \right)^4 \quad (12) \]

The forms loss associated with the jet is similar to the forms loss associated with jetting onto a baffle. This loss coefficient is

\[ K_{\text{baffle}} = 1 \quad (\text{Idelchik, 1994}) \]

The forms loss associated with a 90° turn is
The area associated with region 3 can be calculated as

\[
A_3 = A_{\text{impact}} - A_{3/8''OD}
\]  

(13)

\[
A_3 = \left( \frac{\pi D_{\text{BeamStrike}}^2}{4} + (H - D_{\text{BeamStrike}})D_{\text{BeamStrike}} \right) - \frac{\pi D_{3/8''}^2}{4}
\]  

(14)

The friction factor for smooth piping, as a function of the Reynolds number is

\[ f = a \, \text{Re}^b \]  

(Incropera, 2007)

The pressure balance can be rewritten as

\[
\Delta P = \left\{ \frac{L_p}{D_p} a_p \, \text{Re}^b \right\} \left\{ \frac{G_p^2}{2 \rho g_c} + \left\{ K_n + K_{\text{baffle}} \right\} \frac{G_2^2}{2 \rho g_c} + \left\{ K_{90} \right\} \frac{G_3^2}{2 \rho g_c} \right\}
\]  

(15)

When the mass flux and Reynolds number associated with each section of piping are written in terms of the mass flow rate, the pressure balance is

\[
\Delta P = \left\{ \frac{L_p}{D_p} a_p \left( \frac{D_p}{\mu A_p} \right)^b \right\} \left\{ \frac{m^2}{2 \rho g_c A_p^2} + \left\{ K_n + K_{\text{baffle}} \right\} \frac{m_2^2}{2 \rho g_c A_2^2} + \left\{ K_{90} \right\} \frac{m_3^2}{2 \rho g_c A_3^2} \right\}
\]  

(16)

Equation 16 can be solved implicitly for the mass flow rate.
Chapter 5

Target Chamber Modeling

5.1 Description

Under steady-state operating conditions, the target chamber is expected to be comprised of a boiling region and a condensing region. The boiling region will be located in the bottom of the chamber with height \( H_b \) and above it will be the condensing region with height \( H_{cd} \). These regions compose the total height of the chamber \( H = H_b + H_{cd} \). Due to the self-regulating nature of thermosyphon targets, the water level or boiling height will be a function of the heat input.

![Boiling Region and Condensing Region in Thermosyphon Target Chamber](image)

Figure 18: Boiling Region and Condensing Region in Thermosyphon Target Chamber

For modeling purposes, the water in both regions was assumed to be under saturation conditions at the chamber pressure. This assumption is reasonable for the boiling and condensing dynamics present during operation.
5.2 Boiling Heat Transfer Coefficient

Heat transfer from the boiling region can be modeled using a boiling heat transfer coefficient which depends on the void fraction and the boiling height. The Nusselt number for volumetrically heated pools is often correlated in the form (Wen, 2006)

\[ Nu = c Ra^a \] (1)

Proposed values for the correlation parameters yield

\[ Nu = \begin{cases} 1.54 Ra^{1/4} & Ra \leq 1.865 \cdot 10^{11} \\ 0.0314 Ra^{0.4} & Ra > 1.865 \cdot 10^{11} \end{cases} \] (2)

where the Rayleigh number is defined to be

\[ Ra \equiv \frac{g \alpha H_b^3 \text{Pr}}{\nu_F^2} \] (3)

The Rayleigh number depends on the Prandtl number, which is given by

\[ \text{Pr} \equiv \left( \frac{C_p \mu}{k} \right) \] (4)

The Rayleigh number also depends on the kinematic viscosity, which is given by

\[ \nu_F \equiv \left( \frac{\mu}{\rho} \right) \] (5)

The boiling heat transfer coefficient, as a function of the Nusselt number is

\[ h_{\text{boil}} = \frac{k_F}{H_b} Nu \] (6)

5.3 Condensing Heat Transfer Coefficient

Heat transfer from the condensing region can be modeled using a condensing heat transfer coefficient. A correlation for film condensation on a vertical plate is
\[ h_{CD} = 0.943 \left[ \frac{g_c \rho_f \rho_{fg} k^3 h'_{fg}}{\mu(T_{sat} - T_{wall}) H_{CD}} \right]^{1/4} \]  \hspace{1cm} (7)

where

\[ h'_{fg} = h_{fg} + 0.68 C_p (T_{sat} - T_{wall}) \]  \hspace{1cm} (8)

such that the condensing heat transfer coefficient can be written as

\[ h_{CD} = 0.943 \left[ \frac{g_c \rho_f \rho_{fg} k^3 \left[ h_{fg} + 0.68 C_p (T_{sat} - T_{wall}) \right]}{\mu(T_{sat} - T_{w}) H_{CD}} \right]^{1/4} \]  \hspace{1cm} (9)

(Incropera, 1996)

### 5.4 Void Fraction Calculation

The vapor flow rate leaving the boiling region is a function of the boiling region void fraction and the bubble rise velocity. At steady state, this vapor flow rate is equal to the condensation rate in the condenser.

\[ \dot{m}_{CD}(\alpha) = \alpha \rho_g v_g A_x \]  \hspace{1cm} (10)

The cross-sectional area is the product of the chamber width and the chamber depth,

\[ A_x = W \cdot D \]  \hspace{1cm} (11)

The heat transfer rate out of the condensing region can then be expressed as

\[ \dot{Q}_{CD}(H_{CD}) = \dot{m}_{CD}(\alpha) h_{fg} \]  \hspace{1cm} (12)

where the vapor velocity is given by Wallis (Wallis, 1969).

\[ v_g = C_o j + v_{g_j} \]  \hspace{1cm} (13)

where

\[ C_o = 1.2 \]
\[ v_{g,j} = 1.53 \left( \frac{\sigma g g_c (\rho_f - \rho_g)}{\rho_g} \right)^{1/4} \]  
\tag{14}

\[ j = \alpha_g v_g + \alpha_j v_j \]  
\tag{15}

If we treat the system as bubbles rising in a stagnant liquid, then \( v_j = 0 \) and the vapor velocity reduces to

\[ v_g = C_o \alpha_g v_g + v_{g,j} \]  
\tag{16}

\[ v_g = v_{g,j} (1 - C_o \alpha_g)^{-1} \]  
\tag{17}

Substituting Equation 17 into Equation 12, the heat transfer rate out of the condensing region is

\[ \dot{Q}_{CD} (H_{CD}) = \alpha \rho_g v_g A_s (1 - C_o \alpha)^{-1} h_{jg} \]  
\tag{18}

and solving for the void fraction gives

\[ \frac{\alpha}{1 - C_o \alpha} = \frac{\dot{Q}_{cd} (H_{CD})}{\rho_g v_g A_s h_{jg}} = \xi(H_{CD}) \]  
\tag{19}

\[ \alpha = \frac{\xi(H_{CD})}{1 + C_o \xi(H_{CD})} \]  
\tag{20}

\textbf{5.5 Total Heat Transfer Rate}

Total heat transfer out of the target chamber is the sum of the heat transferred radially out to the cooling channels and the heat transferred out of the back of the target to the jet. This heat transfer is equal to the heat input from the proton beam. The heat balance is

\[ \dot{Q} = \dot{Q}_{\text{boil}} + \dot{Q}_{CD} = \dot{Q}_{\text{boil1}} + \dot{Q}_{\text{boil2}} + \dot{Q}_{CD1} + \dot{Q}_{CD2} \]  
\tag{21}

In Equation 21, subscript 1 represents radial transfer and subscript 2 represents transfer through the back of the target.
The heat transfer through the back of the target can be approximated as one-
dimensional heat transfer through a flat plate, and is given by

\[
\dot{Q}_{\text{boil}2} = \left( \frac{1}{h_{\text{boil}}} + \frac{l}{k} + \frac{1}{h_{\text{jet}}} \right)^{-1} A_{\text{boil}2} (T_{\text{sat}} - T_{\text{cool}}) 
\]

(22)

\[
\dot{Q}_{\text{CD}2} = \left( \frac{1}{h_{\text{CD}2}} + \frac{l}{k} + \frac{1}{h_{\text{jet}}} \right)^{-1} A_{\text{CD}2} (T_{\text{sat}} - T_{\text{cool}}) 
\]

(23)

The heat transfer in the radial direction is given by

\[
\dot{Q}_{\text{boil}1} = \int_{A_{\text{boil}}} h_{\text{boil}} (T_{\text{sat}} - T_{\text{wall}}) dA 
\]

(24)

\[
\dot{Q}_{\text{CD}1} = \int_{A_{\text{CD}1}} h_{\text{CD1}} (T_{\text{sat}} - T_{\text{wall}}) dA 
\]

(25)

These heat transfer rates will need to be evaluated numerically.

The condensing heat transfer rate has been shown to be a function of the wall
temperature and the condensing height. Similarly, the boiling heat transfer coefficient has
been shown to be a function of the void fraction and the boiling height. The sum of the
boiling height and condensing height is known to be the total height of the target chamber.

For known wall temperature, the heat input can be written as

\[
\dot{Q} = \dot{Q}_{\text{boil}} (\alpha, H_{\text{CD}}) + \dot{Q}_{\text{CD}} (H_{\text{CD}}) 
\]

(26)

The void fraction, also a function of the condensing height, is given by

\[
\dot{Q} = \dot{Q}_{\text{boil}} [\alpha(H_{\text{CD}}), H_{\text{CD}}] + \dot{Q}_{\text{CD}} (H_{\text{CD}}) 
\]

(27)

The total heat input, which corresponds to the total heat transfer rate, is shown to be a single
nonlinear equation in the condensing height. This equation can be solved iteratively to yield
condensing height and the total heat transfer rates from the boiling and condensing regions.
Chapter 6

Integrated Model

6.1 Overview

Total heat input has been linked to the condensing height through a single nonlinear equation. A FORTRAN code was written to evaluate this equation using the heat transfer coefficient correlations and physics previously described. The COMSOL Multiphysics program, which uses finite element techniques to solve partial differential equations, was used to model the radial heat transfer. Heat transfer coefficients calculated in the FORTRAN code were input into a COMSOL Multiphysics heat transfer model. The COMSOL solution yielded wall temperatures, boiling heat transfer rates in the radial direction, and condensing heat transfer rates in the radial direction. Iteration between the FORTRAN code and COMSOL model yielded a converged solution for heat input and corresponding condensing height.

Radial temperature profiles for a 15 mm total height target chamber operating at 300 psi pressure were produced using COMSOL Multiphysics. This target chamber size corresponds to 30 radial coolant channels with heat transfer coefficient, $h_{cool} = 7608.339$ Btu/hr-ft$^2$-F.
The converged solution for the 10 mm boiling height thermal limit corresponds to boiling heat transfer coefficient, $h_{\text{boil}} = 1967.532$ Btu/hr-ft²-F, condensing heat transfer coefficient, $h_{\text{cond}} = 2429.482$ Btu/hr-ft²-F, and a heat input of 1394 W.

Figure 19: COMSOL Radial Temperature Profile, 15 mm Height Target at 300 psi at Thermal Limit
The converged solution for a 12.5 mm boiling height corresponds to boiling heat transfer coefficient, $h_{\text{boil}} = 1775.981 \text{ Btu/hr-ft}^2\text{-F}$, condensing heat transfer coefficient, $h_{\text{CDi}} = 2923.994 \text{ Btu/hr-ft}^2\text{-F}$, and a heat input of 1333 W.

Figure 20: COMSOL Radial Temp. Profile, 15 mm Height Target at 300 psi at 12.5 mm Boiling Height
Radial temperature distributions indicate that the coolant channels are very effective at removing the heat that is conducted in the radial direction. Temperatures exceed 120°F at the walls of the target chamber, but are reduced to room temperature beyond the coolant channels. No coolant channels are present directly above or below the target chamber to allow for necessary fill lines. In these regions, temperatures are significantly higher for longer distances into the target body due to longer conduction distances to the nearest coolant channels.

### 6.2 Code Description

The FORTRAN Code allows user input of desired target chamber geometry: the target height, target depth, and the target width as well as the operating pressure. The target width is set by the cyclotron beam configuration. The target depth is set to maintain range thickness for the given proton energy. The target height is adjusted to achieve the desired heat output and maximum void fraction in the boiling region.

The FORTRAN code takes a specified condensing height and determines the corresponding heat input. From the specified geometry, the mass flow rates through the radial coolant channels and the jet cooling system can be determined. These mass flow rates determine the cooling water heat transfer coefficients. An initial guess for the void fraction is required to compute an initial heat transfer coefficient for the boiling and condensing regions. The code outputs the heat transfer coefficients for the radial channels and for the radial boiling and condensing regions. These heat transfer coefficients are input into a COMSOL Multiphysics model of a radial target cross-section similar to those illustrated in Figures 19 and 20. Wall temperature and heat transfer rates are evaluated by COMSOL and re-input into the FORTRAN code. This process is iterated until a converged solution is
obtained. Heat transfer coefficients for the jet cooling system, the back boiling region, and the back condensing region are used to solve the conduction problem through the back plate. The total condensing heat transfer rate from the radial channels and jetting onto the back plate is used to calculate a new void fraction. This process can be repeated using the new void fraction until convergence is reached. When converged, the total heat input corresponding to the specified condensing height and total height has been determined. A flow chart for the iterative solution is given in Figure 21.
6.3 Target Optimization

Simulations were run to evaluate the performance of targets with total heights between 13 mm and 17 mm. For each total height, the boiling height was varied between the thermal limit (10 mm) and the top of the target. Simulations were also run over a range of pressures from atmospheric up to 400 psi. For each simulation the heat input was determined as a function of the boiling height.

![Heat Input vs. Boiling Height](image)

Based on these results, the cyclotron staff at Duke Medical center selected a total target height of 15 mm.
6.4 Results

The performance of the 15 mm height target operating at 300 psi was examined over a range of possible boiling heights. The use of fins on the back of the target to increase heat transfer to the jet was proposed, but neglected in the simulation for conservatism. The figure below illustrates part of the interior target flange. A cross-sectional view of the target chamber is visible, with 10 mm width and 15 mm total height. A cross-section of the 30 radial coolant channels can be seen located at 0.01” conduction distance around the target chamber. No coolant channels are present directly above or below the chamber to allow for necessary fill lines.

![Figure 23: CAD Drawing for Prototype Tantalum Target Internals (Mark Humphrey 2006)](image)

The final dimensions selected for the new target were 10 mm width, 15 mm height, and 15 mm depth. This corresponds to a total chamber volume of 1.93 cc, which is equivalent to 1.93 mL of water. The volume is reasonably small, given the high price of O^{18}-enriched water. The thickness of the target back was chosen to be 0.04”. This thickness was
chosen as a conservative minimum to prevent excessive deformation in the event of an over-
pressurization, i.e. pressures exceeding 1000 psi.

Corresponding heat input and void fraction were determined for a range of boiling heights between the thermal limit (10 mm) and the top of the target. Maximum heat input that would not result in boiling within the target was also determined. For boiling heights extremely close to the top of the chamber, a model which includes separate boiling and condensing regions may not be appropriate. Void fraction should increase monotonically and water level should decrease monotonically with the heat input. As a result, the physical conditions in this region of heat input can be interpolated from the calculated data.

Table 3: Heat Input and Condensing Height for Target with 15 mm Total Height

<table>
<thead>
<tr>
<th>$H_{CD}$ (mm)</th>
<th>$H_{B}$ (mm)</th>
<th>$P_z$ (psi)</th>
<th>$\alpha$</th>
<th>$\dot{Q}_T$ (W)</th>
<th>$\dot{Q}_{B-r}$ (W)</th>
<th>$\dot{Q}_{CD-r}$ (W)</th>
<th>$\dot{Q}_{B-j}$ (W)</th>
<th>$\dot{Q}_{CD-j}$ (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>10</td>
<td>300</td>
<td>0.4106</td>
<td>1394</td>
<td>677</td>
<td>486</td>
<td>152</td>
<td>79</td>
</tr>
<tr>
<td>2.5</td>
<td>12.5</td>
<td>300</td>
<td>0.3407</td>
<td>1333</td>
<td>753</td>
<td>367</td>
<td>178</td>
<td>35</td>
</tr>
<tr>
<td>2</td>
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<td>0.3209</td>
<td>1311</td>
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<td>181</td>
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<tr>
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<td>1284</td>
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<td>0</td>
<td>215</td>
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<td></td>
<td></td>
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</tbody>
</table>

Table 4: Heat Transfer Partitioning for Target with 15 mm Total Height

<table>
<thead>
<tr>
<th>$H_{CD}$ (mm)</th>
<th>$H_{B}$ (mm)</th>
<th>$P_z$ (psi)</th>
<th>$\alpha$</th>
<th>$\dot{Q}_T$ (W)</th>
<th>$\dot{Q}_{rad}$ (W)</th>
<th>$\dot{Q}_T$</th>
<th>$\dot{Q}_{Boil}$ (W)</th>
<th>$\dot{Q}_T$</th>
<th>$\dot{Q}_{CD}$ (W)</th>
<th>$\dot{Q}_T$</th>
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</thead>
<tbody>
<tr>
<td>5</td>
<td>10</td>
<td>300</td>
<td>0.4106</td>
<td>1394</td>
<td>1245</td>
<td>84%</td>
<td>40%</td>
<td>40%</td>
<td>30%</td>
<td>30%</td>
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<tr>
<td>2.5</td>
<td>12.5</td>
<td>300</td>
<td>0.3407</td>
<td>1333</td>
<td>1311</td>
<td>84%</td>
<td>72%</td>
<td>72%</td>
<td>28%</td>
<td>28%</td>
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<td>2</td>
<td>13</td>
<td>300</td>
<td>0.3209</td>
<td>1311</td>
<td>1311</td>
<td>84%</td>
<td>75%</td>
<td>75%</td>
<td>25%</td>
<td>25%</td>
</tr>
<tr>
<td>1.5</td>
<td>13.5</td>
<td>300</td>
<td>0.2970</td>
<td>1284</td>
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<td>84%</td>
<td>78%</td>
<td>78%</td>
<td>22%</td>
<td>22%</td>
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<td>1</td>
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<tr>
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<td>300</td>
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<td>0</td>
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<td>215</td>
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<td>83%</td>
<td>83%</td>
<td>83%</td>
<td>83%</td>
<td>83%</td>
</tr>
</tbody>
</table>

The generated results indicate that roughly 84% of the heat is being removed through the radial channels, with the other 16% removed by the jet cooling system. This illustrates
the value of including radial coolant channels over conventional target designs which only
include cooling via a jet on the target back.

Figure 24: Void Fraction vs. Heat Input for Target with 15 mm Total Height

Figure 25: Boiling Height vs. Heat Input for Target with 15 mm Total Height
In the figures above, the void fraction and boiling height which correspond to thermal limits are indicated in red. The thermal limit corresponds to the minimum heat input that would result in average void fraction in excess of 60% or for water level, i.e. boiling height, below 10 mm. The figures indicate that the Duke target will be limited by the minimum boiling height condition rather than the maximum void condition. The thermal limit, or minimum heat input that would result in beam striking the back wall of the target, is 1394 Watts. This heat input is well above the desired target operating condition, 550 Watts, and the maximum thermal output of the Duke cyclotron, roughly 1 kW, which suggests that this design is sufficiently conservative. Under operating conditions, the target is predicted to have average void fraction less than 0.2187 and water level above 14.5 mm. The figures also show that in a certain heat input range, small changes in heat input can result in large changes in the water level. This behavior indicates the need for conservatism in the target design.

6.5 Sensitivity Analysis

A sensitivity analysis was performed on the FORTRAN/COMSOL model used to generate these results. A relative uncertainty was introduced into key modeling parameters, and the effect on the heat input and thermal limit for a fixed boiling height was examined. A fixed boiling height of 12.5 mm was selected for this sensitivity analysis. This height corresponds to a water level halfway between the thermally limiting height and the top of the chamber.

Mass flow rates through the radial and jet cooling systems were estimated using characteristic pressure forms losses. There is some uncertainty associated with these pressure losses. The only way to reduce this uncertainty will be to measure the mass flow rates experimentally after the target is constructed. The forms losses associated with the
radial coolant channel flow, the jet cooling system flow, and the flow in the cyclotron system piping were each varied by 20%, a factor of 2, and a factor of 10.

<table>
<thead>
<tr>
<th>Sensitivity Parameter</th>
<th>( \delta )</th>
<th>( \alpha )</th>
<th>( \dot{Q}_T ) (W)</th>
<th>( \dot{Q}_{\text{limit}} ) (W)</th>
<th>( \delta \alpha )</th>
<th>( \delta \dot{Q}_T ) (W)</th>
<th>( \delta \dot{Q}_{\text{limit}} ) (W)</th>
</tr>
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<tbody>
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<td>Radial Channels Loss Coefficient ( \delta \cdot (K_{in} + K_{ex}) )</td>
<td>1.2</td>
<td>0.3404</td>
<td>1331</td>
<td>1392</td>
<td>- 0.09%</td>
<td>- 0.13%</td>
<td>- 0.12%</td>
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<tr>
<td></td>
<td>0.8</td>
<td>0.3410</td>
<td>1334</td>
<td>1395</td>
<td>0.09%</td>
<td>0.13%</td>
<td>0.12%</td>
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<td>- 3.76%</td>
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<td>0.1</td>
<td>0.3420</td>
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<td>0.61%</td>
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<td>Jetting Losses ( \delta \cdot (K_{n} + K_{b}) )</td>
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<td>0.23%</td>
<td>0.24%</td>
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<td></td>
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<td>1409</td>
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<td>1.16%</td>
<td>1.11%</td>
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</table>

The results indicate that the target performance as predicted by the model is relatively insensitive to the assumed forms losses. Even when the losses are increased by an order of magnitude, the decrease in the heat input, thermal limit, and void fraction is less than 6% for the cyclotron piping, less than 4% for the radial coolant channels, and less than 2% for the jet system. These results imply that the majority of the pressure loss is due to friction in the straight pipe sections. The piping loss coefficient was set to force the computed pressure losses to match those measured experimentally. As a result, the uncertainty in the total pressure loss is due primarily to uncertainty in the friction loss in the piping inside of the target. Pressure drops across the target have been shown to be small relative to the total
pressure drop for previous thermosyphon targets (Matthew Stokely), so this uncertainty should be fairly small.

Another source for uncertainty is in the heat transfer coefficients. These coefficients are calculated from correlations, based on experimental data. The correlations are formed using dimensionless groups, in order to be applicable over a wide range of geometries. The heat transfer coefficients for jetting heat transfer, condensing heat transfer, and boiling heat transfer were each varied by 20% and a factor of 2. The changes in the heat input and thermal limit for the boiling and condensing heat transfer coefficients were then normalized by their associated surface area for ease of comparison.

\[
\delta\dot{Q}_b' = \delta\dot{Q}_b \cdot \left(\frac{A_b + A_{CD}}{A_b}\right)
\]

<table>
<thead>
<tr>
<th>Sensitivity Parameter</th>
<th>(\delta)</th>
<th>(\alpha)</th>
<th>(\dot{Q}_T) (W)</th>
<th>(\dot{Q}_{\text{limit}}) (W)</th>
<th>(\delta\alpha)</th>
<th>(\delta\dot{Q}_T) (W)</th>
<th>(\delta\dot{Q}_{\text{limit}}) (W)</th>
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<td>1397</td>
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<td>0.22%</td>
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<tr>
<td></td>
<td>0.8</td>
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<td>0.3416</td>
<td>1342</td>
<td>1404</td>
<td>0.26%</td>
<td>0.68%</td>
<td>0.73%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.3392</td>
<td>1316</td>
<td>1376</td>
<td>-0.44%</td>
<td>-1.22%</td>
<td>-1.30%</td>
</tr>
<tr>
<td>Condensing HT Coefficient (\delta \cdot h_{CD})</td>
<td>1.2</td>
<td>0.3661</td>
<td>1399</td>
<td>1478</td>
<td>7.46%</td>
<td>4.98%</td>
<td>6.02%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3094</td>
<td>1256</td>
<td>1300</td>
<td>-9.19%</td>
<td>-5.72%</td>
<td>-6.71%</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>0.4325</td>
<td>1601</td>
<td>1747</td>
<td>26.94%</td>
<td>20.17%</td>
<td>25.32%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.2460</td>
<td>1116</td>
<td>1136</td>
<td>-27.80%</td>
<td>-16.29%</td>
<td>-18.51%</td>
</tr>
<tr>
<td>Boiling HT Coefficient (\delta \cdot h_{\text{boil}})</td>
<td>1.2</td>
<td>0.3405</td>
<td>1477</td>
<td>1519</td>
<td>-0.06%</td>
<td>10.83%</td>
<td>9.01%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3409</td>
<td>1177</td>
<td>1257</td>
<td>0.06%</td>
<td>-11.69%</td>
<td>-9.79%</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>0.3397</td>
<td>1962</td>
<td>1935</td>
<td>-0.29%</td>
<td>47.23%</td>
<td>38.86%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.3413</td>
<td>918</td>
<td>1029</td>
<td>0.18%</td>
<td>-31.08%</td>
<td>-26.17%</td>
</tr>
</tbody>
</table>
Table 7: Surface Area Normalized Sensitivity of Model to Heat Transfer Coefficients

<table>
<thead>
<tr>
<th>Sensitivity Parameter</th>
<th>δ</th>
<th>α</th>
<th>$\dot{Q}_T$ (W)</th>
<th>$\dot{Q}_\text{limit}$ (W)</th>
<th>$\delta\alpha$</th>
<th>$\delta\dot{Q}_T$ (W)</th>
<th>$\delta\dot{Q}_\text{limit}$ (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing HT Coefficient δ $\cdot h_{CD}$</td>
<td>1.2</td>
<td>0.3661</td>
<td>1399</td>
<td>1478</td>
<td>7.46%</td>
<td>19.70%</td>
<td>23.81%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3094</td>
<td>1256</td>
<td>1300</td>
<td>-9.19%</td>
<td>-22.62%</td>
<td>-26.54%</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>0.4325</td>
<td>1601</td>
<td>1747</td>
<td>26.94%</td>
<td>79.77%</td>
<td>100.14%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.2460</td>
<td>1116</td>
<td>1136</td>
<td>-27.80%</td>
<td>-64.43%</td>
<td>-73.21%</td>
</tr>
<tr>
<td>Boiling HT Coefficient δ $\cdot h_{Boil}$</td>
<td>1.2</td>
<td>0.3405</td>
<td>1477</td>
<td>1519</td>
<td>-0.06%</td>
<td>14.50%</td>
<td>12.06%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3409</td>
<td>1177</td>
<td>1257</td>
<td>0.06%</td>
<td>-15.65%</td>
<td>-13.10%</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>0.3397</td>
<td>1962</td>
<td>1935</td>
<td>-0.29%</td>
<td>63.21%</td>
<td>52.01%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.3413</td>
<td>918</td>
<td>1029</td>
<td>0.18%</td>
<td>-41.60%</td>
<td>-35.03%</td>
</tr>
</tbody>
</table>

These results indicate that the target performance as predicted by the model is fairly insensitive to the jetting heat transfer coefficient. The predicted average void fraction is very sensitive to the condensing heat transfer coefficient, but is still well below the thermal limit even with a factor of 2 change in the condensing heat transfer coefficient. The heat input and thermal limit are very sensitive to the boiling and condensing heat transfer coefficients. After normalizing by the surface area ratio, it can be seen that the predicted values are much more sensitive to the condensing coefficient than the boiling coefficient. Even with a factor of 2 change in either of these coefficients, the predicted thermal limit is well above the target operating heat input.
There is some uncertainty in the vapor velocity correlation and in the assumption of bubbling flow through a stagnant volume. The vapor velocity was varied by 20% and a factor of 2.

<table>
<thead>
<tr>
<th>Sensitivity Parameter</th>
<th>δ</th>
<th>α</th>
<th>$\dot{Q}_f$ (W)</th>
<th>$\dot{Q}_\text{limit}$ (W)</th>
<th>$\delta\alpha$</th>
<th>$\delta\dot{Q}_f$ (W)</th>
<th>$\delta\dot{Q}_\text{limit}$ (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.3407</td>
<td>1333</td>
<td>1394</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vapor Velocity Correlation $\delta \cdot v_g$</td>
<td>1.2</td>
<td>0.3047</td>
<td>1312</td>
<td>1378</td>
<td>-10.57%</td>
<td>-1.56%</td>
<td>-1.13%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3864</td>
<td>1356</td>
<td>1411</td>
<td>13.41%</td>
<td>1.79%</td>
<td>1.26%</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>0.2142</td>
<td>1249</td>
<td>1329</td>
<td>-37.13%</td>
<td>-6.30%</td>
<td>-4.66%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.4836</td>
<td>1400</td>
<td>1442</td>
<td>41.94%</td>
<td>5.06%</td>
<td>3.50%</td>
</tr>
</tbody>
</table>

The predicted void fraction is very sensitive to the vapor velocity, as would be expected. However, even with a factor of 2 change in the vapor velocity, the average void fraction is still well below the thermal limit. The heat input and thermal limit are relatively insensitive to changes in the vapor velocity, less than 7% change in heat for a factor of 2 change in vapor velocity.

Sensitivity of the model to several other parameters: thermal conductivity, header pressure difference, and back thickness was studied. The thermal conductivity of tantalum is known with relatively high accuracy. The header pressure difference at the Duke Cyclotron Center has been shown to drift slightly over time. A conservative target back thickness has been selected, but could be further optimized in the future. A decrease in the thickness of the target back would enhance thermal performance by increasing the heat transfer by the jet. However, a decrease in the thickness of the target back would also lower the maximum target pressure that would result in structural failure. There is only a small amount of uncertainty in these parameters, so they were varied by 20% for sensitivity analysis.
Table 9: Sensitivity of Model to Thermal Conductivity, Pressure Header, and Back Thickness

<table>
<thead>
<tr>
<th>Sensitivity Parameter</th>
<th>$\delta$</th>
<th>$\alpha$</th>
<th>$\dot{Q}_T$ (W)</th>
<th>$\dot{Q}_{\text{limit}}$ (W)</th>
<th>$\delta \alpha$</th>
<th>$\delta \dot{Q}_T$ (W)</th>
<th>$\delta \dot{Q}_{\text{limit}}$ (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>0.3407</td>
<td>1333</td>
<td>1394</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>1.2</td>
<td>0.3449</td>
<td>1358</td>
<td>1420</td>
<td>1.23%</td>
<td>1.93%</td>
<td>1.87%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3349</td>
<td>1297</td>
<td>1358</td>
<td>-1.70%</td>
<td>-2.63%</td>
<td>-2.55%</td>
</tr>
<tr>
<td>$\Delta P_{\text{header}}$</td>
<td>1.2</td>
<td>0.3426</td>
<td>1344</td>
<td>1406</td>
<td>0.56%</td>
<td>0.88%</td>
<td>0.86%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3383</td>
<td>1317</td>
<td>1378</td>
<td>-0.70%</td>
<td>-1.13%</td>
<td>-1.10%</td>
</tr>
<tr>
<td>Back Thickness</td>
<td>1.2</td>
<td>0.3401</td>
<td>1326</td>
<td>1387</td>
<td>-0.18%</td>
<td>-0.48%</td>
<td>-0.51%</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.3413</td>
<td>1339</td>
<td>1401</td>
<td>0.18%</td>
<td>0.50%</td>
<td>0.54%</td>
</tr>
</tbody>
</table>

The predicted average void fraction, heat input, and thermal limit were all fairly insensitive, less than 3% change, to small changes in the thermal conductivity, pressure header, and back thickness.
Chapter 7

Heat Flux and Maximum Temperature on the Window Foil

7.1 Boiling Regimes

Protons passing through the target window foil deposit energy, increasing the foil temperature. The foil is effectively insulated on the side opposite the target chamber, such that water within the target chamber provides window cooling. If the target window exceeds the saturation temperature of the target water, pool boiling can occur on the window surface. Pool boiling from a heated surface goes through several distinct regimes: free convection nucleate boiling, transition boiling, and film boiling, as heat flux off the surface is increased. Operation of the thermosyphon target window under nucleate boiling conditions is desirable, because this regime is characterized by high heat transfer coefficients. During nucleate boiling, isolated bubbles form at nucleation sites, often surface defects, and grow on the surface until their buoyancy forces exceed the surface tension. The bubbles are then carried into the fluid stream, and travel upwards either as isolated bubbles or in jets and columns.

There is a maximum heat flux, known as the critical heat flux, under which nucleate boiling can occur. This critical heat flux corresponds to a condition where bubbles blanket much of the surface, and represents the onset of transition boiling. Heat transfer into the vapor film is much less efficient than into the boiling fluid, which can result in dramatic decreases in the heat flux and significantly higher window temperatures (Incropera, 2007).
7.2 Maximum Temperature on Window Foil

Maximum window temperatures will occur once the heat flux on the target window exceeds the critical heat flux and heat transfer transitions to stable film boiling. High foil temperatures could shorten the life of the foil by weakening the material, or could lead to failure of the foil if the melting temperature is exceeded. High temperatures could also sputter the foil material, increasing contamination of the target water. If predicted foil temperatures are too high, target modifications can be considered to reduce the energy deposition or increase the window cooling. Energy deposition can be controlled by reducing foil thickness or using a lighter foil material. Flowing helium gas can be used to cool the outside of the foil to dissipate some of the heat.
Conduction in the window is given by

\[
\frac{d}{dx} \left( k \frac{dT}{dx} \right) + q'''(x) = 0 \tag{1}
\]

The boundary condition on the side where the beam enters, assuming black body radiation, is

\[
k \frac{dT}{dx} \bigg|_{x=0} = \sigma \varepsilon (T_0^4 - T_{\text{amb}}^4) \tag{2}
\]

where \( \sigma = 5.67 \times 10^{-8} \text{ W m}^{-2} \text{K}^{-4} \) and \( \varepsilon \) is Stefan’s Constant and \( \varepsilon \) is the surface emissivity.

The boundary condition on the target chamber side is

\[
-k \frac{dT}{dx} \bigg|_{x=L} = h_c (T_L - T_e) \tag{3}
\]

The conduction equation can be integrated along the conduction length to yield

\[
\int_0^x \frac{d}{dx'} \left( k \frac{dT}{dx'} \right) dx'' + \int_0^x q'''(x') dx'' = 0 \tag{4}
\]

which reduces to

\[
k \frac{dT}{dx} \bigg|_{x=0} - k \frac{dT}{dx} \bigg|_{x=L} + \int_0^x q'''(x') dx'' = 0 \tag{5}
\]

Applying the known left boundary condition, this equation becomes

\[
k \frac{dT}{dx} - \sigma \varepsilon (T_0^4 - T_{\text{amb}}^4) + \int_0^x q'''(x') dx'' = 0 \tag{6}
\]

Equation 6 can be evaluated at the right boundary of the foil to yield

\[
-h_c (T_L - T_e) - \sigma \varepsilon (T_0^4 - T_{\text{amb}}^4) + \int_0^L q'''(x') dx'' = 0 \tag{7}
\]
Equation 6 can be integrated over the conduction length and evaluated at the right boundary of the foil to yield

\[
\int_0^L k \frac{dT}{dx'} dx' + \int_0^L \sigma \varepsilon (T_0^4 - T_{amb}^4) dx' + \int_0^L \int_0^{x'} q'''(x') dx'' dx' = 0
\]  

(8)

This reduces to

\[
T_L - T_0 - \frac{\sigma \varepsilon (T_0^4 - T_{amb}^4) L}{k} + \frac{1}{k} \int_0^{x'} q'''(x') dx'' dx' = 0
\]  

(9)

Equation 9 can be evaluated at any arbitrary location in the foil to yield

\[
T(x) - T_0 - \frac{x \sigma \varepsilon (T_0^4 - T_{amb}^4)}{k} + \frac{1}{k} \int_0^{x'} q'''(x') dx'' dx' = 0
\]  

(10)

At the location of the maximum temperature, Equation 10 yields

\[
T_{\text{max}} - T_0 - \frac{x_{\text{max}} \sigma \varepsilon (T_0^4 - T_{amb}^4)}{k} + \frac{1}{k} \int_0^{x_{\text{max}}} q'''(x') dx'' dx' = 0
\]  

(11)

The maximum temperature corresponds to the location where the temperature derivative equals zero. From Equation 6, it is shown that \( x_{\text{max}} \) satisfies

\[- \sigma \varepsilon (T_0^4 - T_{amb}^4) + \int_0^{x_{\text{max}}} q'''(x') dx'' = 0
\]  

(12)

The convective heat transfer coefficient depends on the pool boiling regime, and may be a function of the wall temperature \( T_L \). In general, Equations 7 and 9 are a system of two nonlinear equations with two unknowns, \( T_0 \) and \( T_L \), and can be solved iteratively. When the temperature on the left boundary is known, Equation 12 can be solved for the location of the maximum foil temperature. Equation 11 can then be solved for the maximum foil temperature.
7.3 Critical Heat Flux Correlation

Correlations have been developed for predicting critical heat flux for pool boiling on vertical surfaces. For a vertical plate with one side insulated, a critical heat flux correlation is

$$q''_{\text{crit}} = \begin{cases} 0.9 & 5.86 < Bo \\ \frac{1.4}{(H')^{1/4}} & 0.15 < Bo < 5.86 \end{cases}$$

(Lienhard, 1973)

The Bond number, a ratio of the gravitational forces to the surface tension, is given by

$$Bo \equiv \frac{g(\rho_f - \rho_g) L^2}{\rho c_f \sigma}$$

(13)

The Zuber-Kutateladze correlation for critical heat flux on a flat plate is

$$q''_{\text{max}} = \frac{\pi}{24} \frac{h_{fg} \rho_g^{1/2}}{\sigma} \left[ \frac{g \sigma g_c}{\rho_f} \right]^{1/4}$$

(14)

Critical heat flux can be calculated for saturated water conditions at pressures between 300 psi and 500 psi.

<table>
<thead>
<tr>
<th>Table 10: Critical Heat Flux</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure (psi)</td>
</tr>
<tr>
<td>Bond Number</td>
</tr>
<tr>
<td>$q''_{\text{max}}$ (Btu/hr-ft²)</td>
</tr>
<tr>
<td>$q''_{\text{crit}}$ (Btu/hr-ft²)</td>
</tr>
<tr>
<td>$q''_{\text{crit}}$ (W/cm²)</td>
</tr>
</tbody>
</table>

7.4 Film Boiling Heat Transfer Coefficient

If the maximum heat flux on the target window exceeds the critical heat flux, the heat transfer coefficient can be calculated from a film boiling correlation. For maximum heat flux below the critical heat flux, a nucleate boiling heat transfer coefficient is appropriate. A heat
transfer coefficient correlation for film boiling on a plane vertical surface is given by (Mills, 1999)
\[
h_{film} = 0.71 \left[ \frac{\rho_f (\rho_f - \rho_g) g h_{fg}^* k_g^*}{\mu_g L (T_W - T_{sat})} \right]^{3/4}
\]

where
\[
h_{fg}^* = h_{fg} + 0.35 C_p (T_{wall} - T_{sat})
\]
The vapor properties are evaluated at the average film temperature
\[
T_f = \frac{T_{sat} + T_{wall}}{2}
\]
and the liquid properties at \( T_{sat} \).

Under nucleate boiling conditions, the Jens-Lottes correlation can be used to determine the foil surface temperature (Lekakh, 1995).
\[
T_{wall} = T_{sat} + 1.897 q^{1/4} e^{-P/900}
\]
where \( P \) is in psi.

### 7.5 Heat Generation in the Window Foil

Heat is generated in the window foil by the energy deposition of protons in the accelerated beam. Energy deposited in the foil can be transferred into the target pool or radiated off of the front of the target. Radiative heat transfer into air is much less efficient than heat transfer into the boiling pool, so all of the energy deposited into the foil can be assumed to enter the target pool. The heat flux into the boiling pool off of the window foil then equals the energy deposited in the foil per unit area.

Energy deposition in the window foil of protons with energies between 8 MeV and 30 MeV was calculated using SRIM and verified using Monte Carlo N-Particle Extended
The beam shape was assumed to be circular with radius 0.4 cm and area 0.50 cm². An average beam flux was taken to be \(1.24 \cdot 10^{13}\) protons/cm²-s/μA. Energy deposition and corresponding volumetric heat generation rate in the window foil were calculated per μA of beam current for 0.001” thick foils of Tantalum and Havar (Matthew Stokely 2006).

### Table 11: Heat Generation in Tantalum Foil (Matthew Stokely 2006)

<table>
<thead>
<tr>
<th>(E_{\text{beam}}) (MeV)</th>
<th>(\frac{\partial E}{\partial x}) (MeV/mm)</th>
<th>(\frac{q}{\mu A}) (W/μA)</th>
<th>(\frac{q''_{\text{avg}}}{I}) (W/cm²·μA)</th>
<th>(\frac{q'''_{\text{avg}}}{I}) (W/cm³·μA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>14.465908</td>
<td>0.367434</td>
<td>0.734868</td>
<td>289.31816</td>
</tr>
<tr>
<td>25</td>
<td>16.476933</td>
<td>0.418514</td>
<td>0.837028</td>
<td>329.53866</td>
</tr>
<tr>
<td>16</td>
<td>22.54023</td>
<td>0.572522</td>
<td>1.145044</td>
<td>450.8046</td>
</tr>
<tr>
<td>12</td>
<td>27.42312</td>
<td>0.696547</td>
<td>1.393094</td>
<td>548.4624</td>
</tr>
<tr>
<td>8</td>
<td>35.81858</td>
<td>0.909792</td>
<td>1.819584</td>
<td>716.3716</td>
</tr>
</tbody>
</table>

### Table 12: Heat Generation in Havar Foil (Matthew Stokely 2006)

<table>
<thead>
<tr>
<th>(E_{\text{beam}}) (MeV)</th>
<th>(\frac{\partial E}{\partial x}) (MeV/mm)</th>
<th>(\frac{q}{\mu A}) (W/μA)</th>
<th>(\frac{q''_{\text{avg}}}{I}) (W/cm²·μA)</th>
<th>(\frac{q'''_{\text{avg}}}{I}) (W/cm³·μA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>9.922322</td>
<td>0.252027</td>
<td>0.504054</td>
<td>198.44644</td>
</tr>
<tr>
<td>25</td>
<td>11.405086</td>
<td>0.289689</td>
<td>0.579378</td>
<td>228.10172</td>
</tr>
<tr>
<td>16</td>
<td>16.017561</td>
<td>0.406846</td>
<td>0.813692</td>
<td>320.35122</td>
</tr>
<tr>
<td>12</td>
<td>19.86975</td>
<td>0.504692</td>
<td>1.009384</td>
<td>397.395</td>
</tr>
<tr>
<td>8</td>
<td>26.77392</td>
<td>0.680058</td>
<td>1.360116</td>
<td>535.4784</td>
</tr>
</tbody>
</table>

Energy deposition and volumetric heat generation are inversely related to the incident energy of the proton. Foil structural damage can occur when temperatures exceed the material’s softening temperature, generally taken to be around 40% of the melting point (Frost, 1982).

### Table 13: Softening Temperature for Tantalum and Havar

<table>
<thead>
<tr>
<th>Material</th>
<th>Melting Temperature (K)</th>
<th>Softening Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tantalum</td>
<td>3290</td>
<td>1316 (1909.13°F)</td>
</tr>
<tr>
<td>Havar</td>
<td>1753</td>
<td>701.2 (802.49°F)</td>
</tr>
</tbody>
</table>
7.6 Beam Characterization

The Duke Cyclotron beam shape has been characterized experimentally. Beam shape in the x-axis and beam shape in the y-axis can each be approximated using a cosine function (Clark, 2004).

\[
R_{pta} = \frac{1}{4/\pi^2} = \frac{\pi^2}{4} = 2.4674
\]

A peak to average ratio of the beam flux can be determined when the beam optics are known. This peak to average ratio is used to compute maximum volumetric heat generation rate

\[
q'''_{\text{max}} = q'''_{\text{avg}} \cdot R_{pta}
\]  

(18)

and the peak heat flux

\[
q''_{\text{max}} = q''_{\text{avg}} \cdot R_{pta}
\]  

(19)
Peak heat flux for the Duke target will increase linearly with beam intensity, and is shown for beam intensity between 25\(\mu\text{A}\) and 50\(\mu\text{A}\), the expected operating conditions and the maximum cyclotron capacity.

![Peak Heat Flux on Window Foil](image)

**Figure 28: Peak Heat Flux on Window Foil for Duke Target**

For the Duke target, the peak heat flux is well below the critical heat flux for both foil materials for the entire range of beam intensity. Neglecting radiation off the front of the target and assuming a uniform volumetric heat generation rate, the peak foil temperature can be estimated as

\[
T_0 = T_{\text{wall-max}} + \frac{q''''_{\text{max}} L^2}{2k}
\]

where the wall temperature is evaluated using the Jens-Lottes correlation.

\[
T_{\text{wall-max}} = T_{\text{sat}} + 1.897 q''_{\text{max}} \frac{1}{4} e^{P/900}
\]
Figure 29: Maximum Temperature on Tantalum Window Foil for Duke Target
For the Duke target, peak window temperature is expected to remain below the softening temperature for both tantalum and Havar foils. The margin for tantalum is much greater than for Havar. Targets with higher beam intensity and/or lower incident proton energy would have higher peak window temperatures. This may be limiting for the 2.4 kW Wisconsin Medical Cyclotron target.
Chapter 8

Conclusions

8.1 Summary and Conclusions

The purpose of this project was to develop methods that may be used to design new thermosyphon water targets for production of $^{18}$F isotope for use in PET medical imaging. These methods would then be used to design a target that would accommodate the highest heat input with the minimum liquid volume without exceeding thermal limits. These thermal limits can be equated to a maximum void fraction in the boiling region, minimum boiling height or overheating of the target window. A FORTRAN/COMSOL Multiphysics code system was developed to characterize boiling conditions in the target under specified heat input conditions.

The design methods and corresponding code system developed were used to design an optimized thermosyphon target for the cyclotron at the Duke University Medical Center in Durham, NC. The Duke target required a 10 mm target chamber width and was intended to operate at 300 psi with 22 MeV particles at 25 $\mu$A beam current. The thermal limit for the Duke target is the minimum heat input that would produce average void fraction above 60% or boiling height below 10 mm.

The optimum target chamber height was determined to be 15 mm. The target was shown to be limited by the minimum boiling height condition rather than the maximum void condition, corresponding to a heat input of roughly 1400 W. This heat input was well above the desired operating heat input of 550 Watts, which suggests that this design is sufficiently
conservative. Under steady-state operating conditions, the target was predicted to have average void fraction less than 0.2187 and water level above 14.5 mm. It was also shown that in a certain heat input range, small changes in heat input can result in large changes in the water level. This behavior supports the large degree of conservatism in the target design.

Sensitivity analysis was performed on the FOTRAN/COMSOL code system for a 15 mm height target. A relative uncertainty was introduced into several key parameters, and the effect on the heat input and thermal limit for a fixed boiling height of 12.5 mm was examined. The model was shown to be most sensitive to uncertainties in the boiling and condensing heat transfer coefficients.

8.2 Future Work

Work is currently underway to develop improved methods to characterize the beam optics. Beam characterization will allow for better control over the heat input to the target and improved estimation of critical heat flux. This information can be used to assess any potential need for cooling the front window foil.

The results of this study have been used to design a new thermosyphon target for use at the Duke Medical Center cyclotron. Construction of this new target is underway. Once constructed, the new thermosyphon target will be tested at the Duke facility. This experimental testing can be used to validate the design code developed in this study.
Figure 31: Axial Cross-Section of 15 mm Height Duke Target (Mark Humphrey 2006)
Figure 32: Radial Cross-Section of 15 mm Height Duke Target (Mark Humphrey 2006)
Figure 33: Cross-Section of Jet Impingement Cooling System for 15 mm Height Duke Target
(Mark Humphrey 2006)

Figure 34: Close-Up of Cross-Section of Jet Impingement Cooling System for 15 mm Height Duke Target
(Mark Humphrey 2006)
The design methods and corresponding code system developed in this study will also be used to design a thermosyphon target for the Wisconsin Medical Cyclotron (WMC). The WMC accelerates protons to 16.5 MeV and can operate with 150 μA beam current. This corresponds to a heat input of roughly 2.4 kW. Because this heat input is higher than conventional boiling water targets have been able to support, design and testing of a WMC thermosyphon target will be of great interest in the cyclotron target community.
List of References


