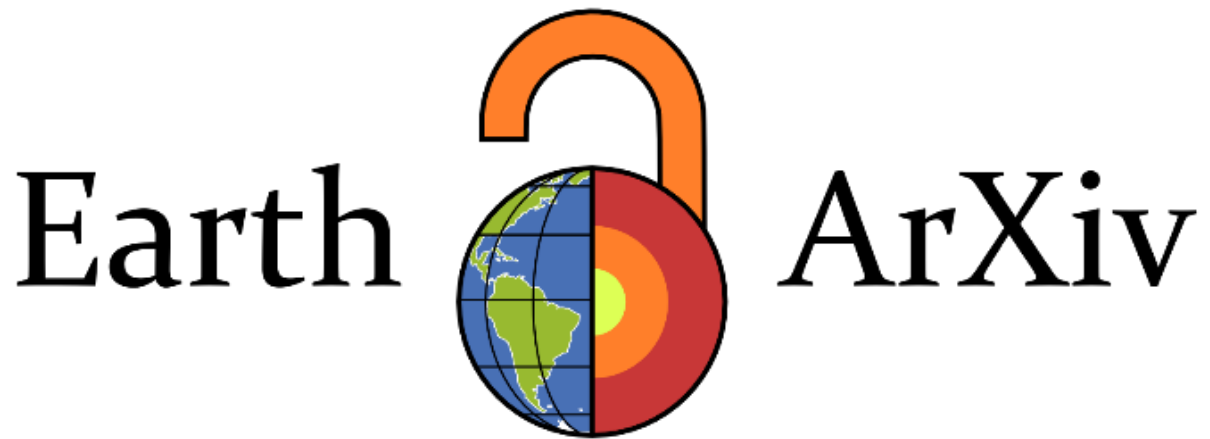




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Models of Thermosyphon Performance, Design, and
Influence on Glacier Flow

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ABSTRACT. Sea-level rise threatens coastal communities worldwide, and the acceleration of West Antarctic glaciers is the largest source of uncertainty in sea-level projections. The melt rate at the ice-bed interface is a primary control on the flow speed of fast-moving Antarctic glaciers and ice streams, whose beds are lubricated by meltwater and governed by thermodynamic feedback. Thermosyphons are passive heat-transfer devices, common in Arctic permafrost stabilization, that transport heat vertically through a phase-changing working fluid. By cooling the basal interface, thermosyphons could reduce meltwater production and slow glacial flow by increasing basal drag. Here, we present a research roadmap to evaluate thermosyphons as a tool to remediate climate-driven acceleration by slowing Antarctic glaciers. We develop a thermal model that couples conductive heat transfer through the glacier with thermosyphon performance metrics. We find that thermosyphons can extract basal heat on the same order of magnitude as natural geothermal flux. When we incorporate the thermal model into an ice-sheet model, these perturbations to basal heat flux have the potential to significantly reduce ice-stream velocity. These results suggest thermosyphons warrant further investigation as a potential remediation strategy for stabilizing Antarctic glaciers to minimize the likelihood of rapid rates of sea-level rise.

1. INTRODUCTION AND BACKGROUND

1.1. Background

Sea level rise requires adaptation and migration in coastal communities (Hauer and others, 2020; Hinkel and others, 2014, 2018; Appeaning Addo, 2015; Fang and others, 2020; Edwards and others, 2021). The acceleration and collapse of West Antarctic glaciers is the largest source of sea level uncertainty in the 21st century (Edwards and others, 2021; Coulon and others, 2024). In some high-range scenarios, mass loss from the West Antarctic Ice Sheet alone is expected to contribute on the order of 50 cm to sea level rise, which

would expose up to 80 million people to flooding by 2100 (Bamber and others, 2019; Kulp and Strauss, 2019; McMichael and others, 2020; Oppenheimer, M. and others, 2022). Anthropogenic climate change, resulting in warming ocean waters near Antarctica and elevated surface temperatures has been shown to significantly increase the trajectory of mass loss from West Antarctica (Bulthuis and others, 2019).

Ice streams are regional areas of relatively fast ice flow; on the Siple Coast, some ice streams are actively flowing, whereas the Kamb Ice Stream has stopped (Retzlaff and Bentley, 1993; Ng and Conway, 2004; Catania and others, 2006) and the Whillans Ice Stream is slowing down (Joughin and Tulaczyk, 2002). Observations with ice penetrating radar suggest that the Kamb Ice Stream has numerous buried crevasses covered by surface snow, which indicates that it was previously flowing, and that it stagnated and froze to the bed approximately 100-200 years ago (Retzlaff and Bentley, 1993; Ng and Conway, 2004; Catania and others, 2006). In addition to the stagnation of the Kamb Ice Stream, Whillans Ice Stream is slowing down from similar processes, likely in the process of stagnating (Joughin and Tulaczyk, 2002). The periodic flow and stagnation of these ice streams can be described by a thermal model of the meltwater system at the bed which is then linked into a dynamical model to find stability conditions (Tulaczyk and others, 2000; Robel and others, 2013). Climate variability has a significant effect on ice sheet dynamics due to these thermodynamic feedback processes (Robel and others, 2013; Mantelli and others, 2016; Robel and Tziperman, 2016). Predicting and understanding the speed and variability of ice stream flow and stagnation is critical to understanding future sea level rise from Antarctica.

Understanding sea level rise and addressing its impacts remains a critical challenge and reducing anthropogenic carbon emissions is vital towards minimizing further increase in surface temperature. Ice sheets are being damaged by a warming climate and some areas of the ice sheet could be lost to future generations if they continue to devolve. Glacial remediation through ice sheet intervention could be a means to minimize the damage to ice sheets caused by a warming climate. The decision to employ remediation strategies is a complex one involving many stakeholders and communities, and this research is an early step towards understanding glacier remediation options while offering an opportunity to learn more about the glacier system.

With the goal of reducing sea level rise, three categories of interventions have been proposed. Our focus is “basal” techniques which intervene at the interface between the glacier and the bed (Meyer and others, 2026; Lockley and others, 2020; MacAyeal, 1983) rather than “surface” techniques which change processes on the surface of the glacier (MacAyeal, 1983; Field and others, 2018), or “ocean” techniques which change the interface between the ocean and ice shelves (Cathcart and others, 2011; Wolovick and others, 2023; Wolovick and Moore, 2018). Known basal intervention techniques include cooling the bed using passive thermosyphons (Lockley and others, 2020), pumping water from the bed (Meyer and others, 2026), and directly increasing basal drag (Lockley and others, 2020). Basal interventions are of particular interest because they utilize natural dynamical feedback within the glacier to drive remediation with minimal intervention and are a potential path to breaking the positive buoyancy-driven feedback, broadly defined as the marine ice sheet instability (MISI), that can drive rapid sea-level rise (Schoof, 2007). In this way, basal interventions are a potential path towards stabilizing the West Antarctic Ice Sheet, which amounts to reducing the sensitivity of the ice sheet to future warming. In the absence of intervention, even if climate forcing on the ice sheet were reduced, internal buoyancy-driven feedback is expected to continue to

accelerate ice loss. Additionally, basal techniques at the interface between the glacier and the bed have no first-order couplings with the ocean or the atmosphere, thus making modeling more tractable and outcomes easier to predict.

Thermosyphons are passive heat-transfer devices, common in Arctic permafrost stabilization (Wagner, 2014), which use phase change in a working fluid to drive heat vertically. Thermosyphons could be used to reduce basal melt in fast-flowing areas of West Antarctica, nudging the glacier to stagnation through thermodynamic feedback. Of the proposed basal intervention techniques, thermosyphons are of interest because they are passive, and can be easily shut off, remotely if necessary, by extracting the internal working fluid. Evaluating the safety and efficacy of thermosyphons involves developing models of thermosyphons themselves, models of the glacier system and working with interconnected human systems and policy landscapes. While a few of the areas of research required to evaluate thermosyphon safety and efficacy are specific to thermosyphons themselves, others are in alignment with existing areas of focus in glaciology, and many are also in common with research that would be needed to investigate other basal interventions. A breakdown of steps to evaluate thermosyphon safety and efficacy is shown in Figure 1, with many of the initial steps taken in this paper.

In this paper, we first discuss basal interventions in the context of ice dynamics and introduce the proposed uses of thermosyphons in glaciers. We then develop a thermal model that analytically represents conductive heat transfer through the glacier, coupled with thermosyphon performance metrics. Finally, we incorporate thermosyphons into an ice-stream flow model of Thwaites Glacier to characterize their potential effect.

1.2. Ice Sheet Remediation through Basal Intervention

Basal interventions are those that alter the physical processes at the base of the glacier with the goal of causing a glacier or ice stream to slow by increasing the drag at the ice-bed interface. At the bed of an Antarctic ice stream, melt rate is governed by a balance of frictional heating, geothermal heat flux and the heat conducted upwards through the glacier. The temperature at the bed is assumed to be at the pressure melting point, near 0 °C, and the average annual surface temperature varies from -30 to -10 °C. The melt rate is then,

$$\dot{m} = \frac{1}{L_{\text{ice}}\rho_{\text{ice}}} [\tau_b U_b + G - k_{\text{ice}}\theta'_b - q_{\text{thermosyphon}}], \quad (1)$$

where τ_b is the shear stress at the bed, U_b is the bed sliding velocity, G is the geothermal heat flux, k_{ice} is the thermal conductivity of ice, $\theta'_b = \frac{dT}{dz}$ is the vertical temperature gradient at the base of the ice, ρ_{ice} is the density of ice, L_{ice} is the latent heat of ice (Tulaczyk and others, 2000). The term $q_{\text{thermosyphon}}$ represents the heat extracted using thermosyphons for basal intervention. Under the assumption that the deformation of the bed till itself is the primary mechanism of slip in ice streams, rather than the form drag of the ice deforming around bed obstacles (Zoet and Iverson, 2020; Minchew and Joughin, 2020), the shear stress at the bed is equal to the yield stress of the till,

$$\tau_b = \tau_{\text{yield,till}} = \mu(\rho_{\text{ice}}gH - p), \quad (2)$$

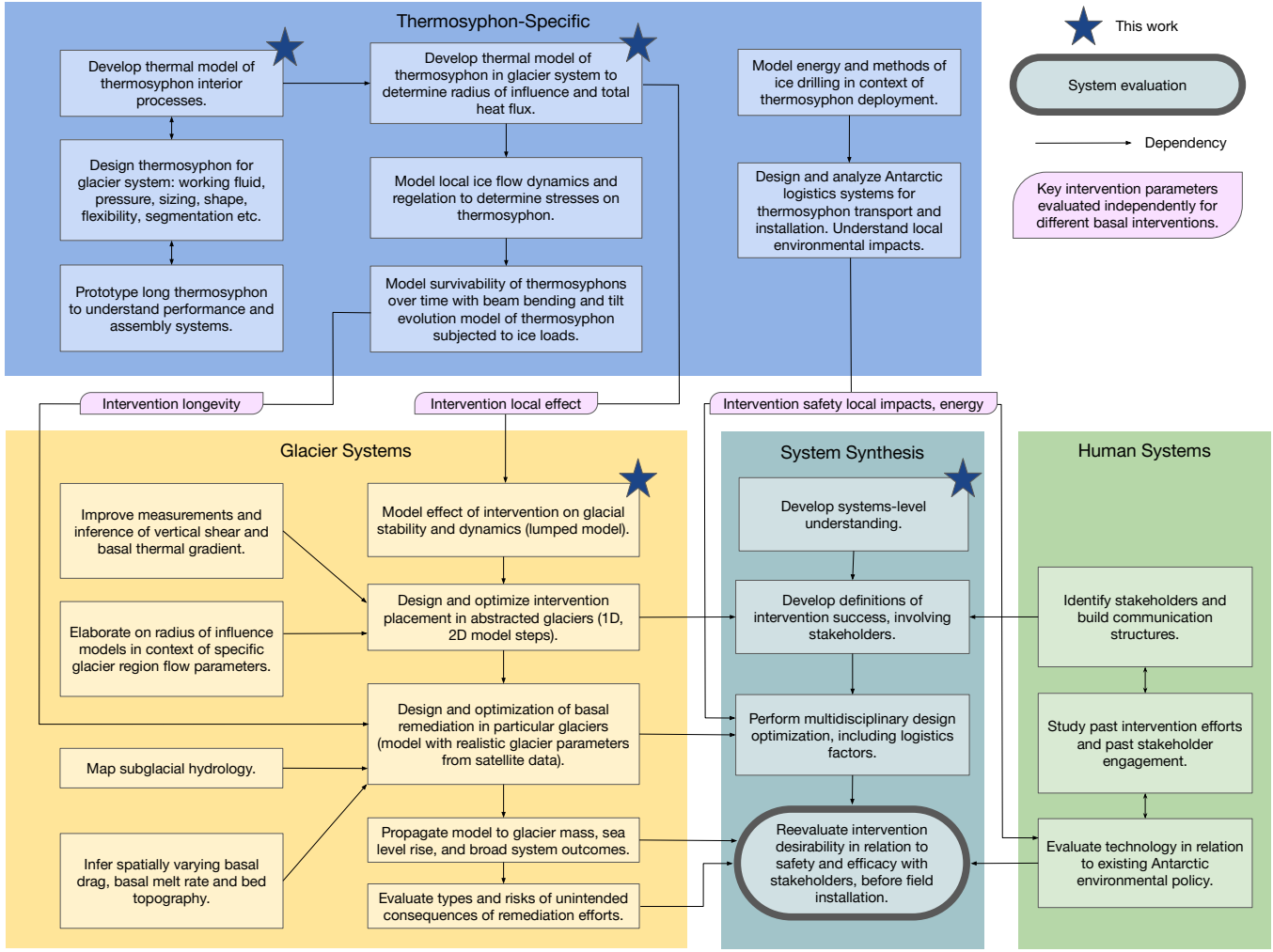


Fig. 1. Breakdown of key research areas in assessing the safety and efficacy of thermosyphons as a glacial climate remediation. Subproblems that are covered in this work are denoted with a star. Areas of research are categorized into those that are specific to thermosyphons, involving glacier systems, human systems, and those that are a combination of all system parts. Note that intervention research areas that are specific to thermosyphons have analogous outputs that are relevant to other basal interventions and allow various basal interventions to be evaluated interchangeably.

where $\mu \approx 0.5$ is the friction coefficient in the deforming till layer (Iverson, 2010), g is the gravitational acceleration, and p is the water pressure in the deforming till layer, and where we ignore the effects of cohesion (Iverson and others, 1998; Iverson, 2010). The bed sliding velocity here is given by the following expression,

$$U_b = \left(\frac{2^{1-n} B_{\text{ice}}^n}{n+1} \right) W^{n+1} H (\tau_d - \tau_b)^n \quad (3)$$

where $\tau_d = -\rho_{\text{ice}} g H \frac{\partial H}{\partial x}$ is the gravitational driving stress, n is the viscous stress exponent for ice, B_{ice} is the coefficient of ice viscosity, and W is the half-width of the ice stream in multiples of ice thickness (Tulaczyk and others, 2000).

From these expressions, one can devise several ways of changing basal flow conditions in a way that results in slowing a glacier by increasing the shear strength of the bed and, in extreme circumstances, freezing the glacier to the bed. For instance, the drag at the bed, τ_b can be increased directly by adding

obstructions at the bed, such as a driven metal pile. Alternatively, the water pressure, p , could be decreased by pumping the water continuously to the surface (Meyer and others, 2026). Another option would be to deposit a large amount of mass on the surface of the glacier, indirectly increasing τ_b (Feldmann and others, 2019). Alternatively, the thermal balance at the bed could be changed by adding cooling devices, essentially changing k (Lockley and others, 2020), or by adding a substance at the bed which freezes at a lower temperature than water. This paper studies the use of passive thermosyphons to cool the bed.

The impact of adding thermosyphons into a glacier can be seen by analyzing the dynamics of Equation (1), Equation (2) and Equation (3) from Tulaczyk and others (2000), with an additional term, $q_{\text{thermosyphon}}$ added to the expression for melt rate in Equation (1), which models the heat flux from thermosyphons. Since $q_{\text{thermosyphon}}$ and G are both independent of τ_b , changing either of them will have the same type of dynamic effect on the system, changing the locations of system stability states, and varying the conditions under which the glacier is nudged to stagnation. Assembling these system dynamics, the melt rate at the bed, $\dot{m}$, varies as a function of basal shear stress, τ_b in as shown in Figure 2, using the parameters from Tulaczyk and others (2000), $\theta'_b = 0.04 \text{ K m}^{-1}$, $\tau_d = 13 \text{ kPa}$, $W = 17 \text{ km}$, $H = 1 \text{ km}$, $n = 3$, $2^{1-n}(n+1)^{-1}B_{\text{ice}}^n = 1.45 \times 10^{-16} \text{ s}^{-1} \text{ kPa}^{-3}$, and values of ρ_{ice} and k_{ice} listed in Table 1. Under conditions where

$$G - q_{\text{thermosyphon}} - k_{\text{ice}}\theta'_b \leq 0 \quad \text{and} \quad G - q_{\text{thermosyphon}} - k_{\text{ice}}\theta'_b \geq -(\tau_b U_b)_{\text{max}}, \quad (4)$$

is satisfied, where $(\tau_b U_b)_{\text{max}} = n^n(n+1)^{-(n+1)}W^{n+1}U_d\tau_d$, there will exist a stable equilibrium where $\dot{m} = 0$ representing constant ice stream flow, and an unstable equilibrium where $\dot{m} = 0$ (Tulaczyk and others, 2000). States with values of τ_b below the unstable equilibrium will go to the stable streaming equilibrium, and values above the unstable equilibrium will trend towards ice stream stagnation. In this part of the state space, adding thermosyphons effectively increases the drag τ_b above which the ice stream will tend towards stagnation. Thermosyphons shift the unstable steady state to the left (by cooling) while simultaneously increasing (moving to the right) the drag (bed strength) by reducing melt rates and thus water pressure. This means thermosyphons could cause a convergence of the glacier basal drag state and the unstable steady state that leads to runaway stagnation. In conditions where

$$G - q_{\text{thermosyphon}} - k_{\text{ice}}\theta'_b < -(\tau_b U_b)_{\text{max}}, \quad (5)$$

there will be no points where $\dot{m} = 0$, and the system will always trend towards stagnation (Tulaczyk and others, 2000). If adding thermosyphons pushes the system into this part of the state space, the ice stream will stagnate for all values of τ_b .

In Antarctic glaciers, melt rate at the bed is subject to high spatial variability, where melt rates in the West Antarctic are commonly in the range of -2 mm yr^{-1} to 20 mm yr^{-1} . Melt rates are closely linked to ice velocity by Equation (1), with velocity for all of Antarctica pictured in Figure 3a (Rignot and others, 2011, 2017; Mouginot and others, 2012). A depiction of this spatial variability of melt rates across Antarctica is shown in Figure 3c, modified from Raspoet and Pattyn (2025). Most areas of Antarctica have melt rates near equilibrium, except those with high velocity such as Thwaites Glacier in Figure 3b, which has basal melt predominantly driven by frictional heating as shown in Figure 3d. In these fast moving areas frictional heating is responsible for nearly all of the basal melt, with frictional melt rate values ranging

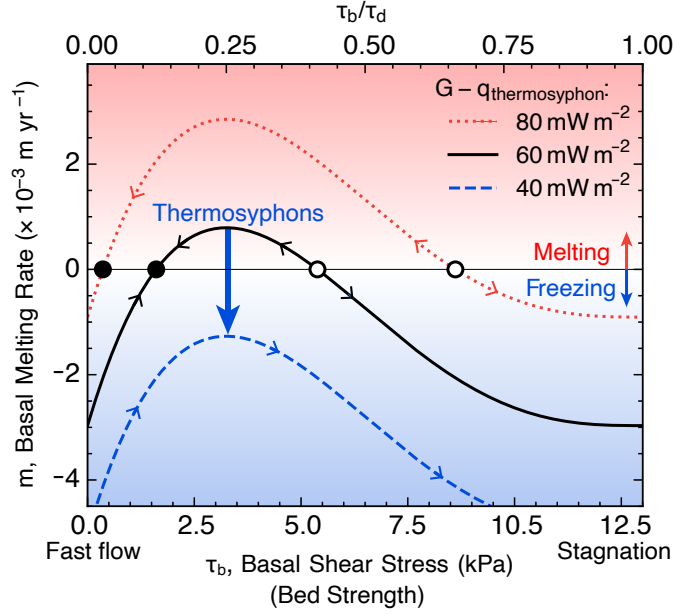


Fig. 2. Thermosyphons act to offset geothermal heat flux in the basal melt system, changing the stability criteria of ice streams in an undrained plastic bed model and causing them to stagnate. Melt rate varies as a function of basal shear stress, exhibiting a stable steady state (closed circle) in a fast flowing ice stream configuration, and an unstable configuration (open circle) where melt rate is zero. Figure modified from Tulaczyk and others (2000).

from 0 mm yr^{-1} to $>1000 \text{ mm yr}^{-1}$ in the fastest-moving sections. As shown by the system dynamics model (Tulaczyk and others, 2000), even small perturbations to the melt rate can cause significant difference in the location of stable states. This paper presents a simple thermosyphon model to find a realistically achievable perturbation to the basal melt rate.

1.3. Thermosyphons and Heat Pipes

Thermosyphons are passive devices for efficient transfer of heat upwards using a phase change in an enclosed convecting working fluid. The bottom of the thermosyphon is filled with the working fluid in a liquid phase, which heats up, transitions to gaseous phase, decreases in density and rises to the top of the thermosyphon. At the top of the thermosyphon, the fluid re-condenses to liquid phase, and flows down the thermosyphon. Thermosyphons rely on gravity to ensure fluid transport and can be operated vertically either in a pipe design or in a loop design. Passive heat transfer devices using phase change can also be built in a horizontal configuration by using a wick or other special structures for material transport. Since thermosyphons rely on the internal working fluid for heat transfer, thermosyphons may be quickly “switched-off” by extracting the working fluid (Mantelli, 2021).

Thermosyphons vary in scale between devices for cooling computer systems that are several centimeters in length to large structures over a kilometer long for geotechnical applications (Mantelli, 2021). Thermosyphons may be used with almost any working fluid that is containable within realistic pressures, and which doesn’t corrode the walls of the container; appropriate working fluids exist for temperatures between 5 K and 1450 K (Mantelli, 2021).

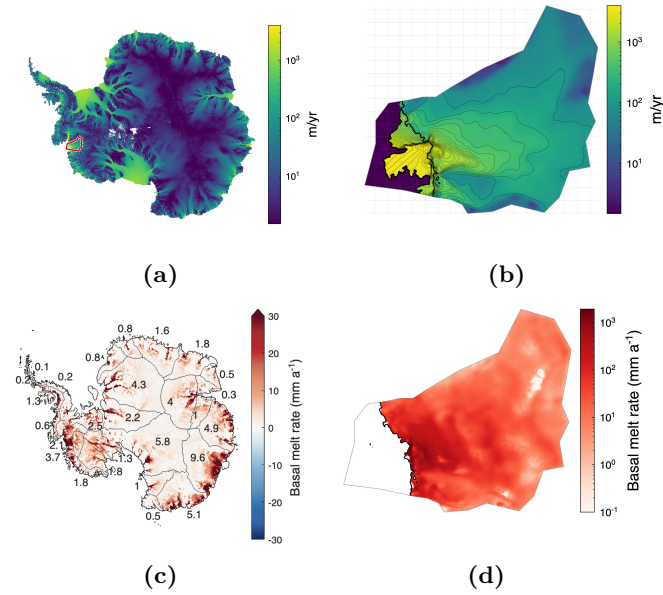


Fig. 3. (a) Ice surface velocity in Antarctica with Thwaites Glacier marked (Rignot and others, 2011, 2017; Mouginot and others, 2012). (b) Ice surface velocity on Thwaites Glacier using domain from Goldberg and others (2026). (c) Basal melt rate values from an ensemble of models across Antarctica (Raspoet and Pattyn, 2025) exhibit a high degree of spatial variability. Averages for various marked geographic areas are denoted. Values higher than 30 mm yr^{-1} were truncated. (d) Basal melt rate due to frictional heating calculated for Thwaites Glacier in Section 3 using an ISSM model from Goldberg and others (2026), showing high spatial variability. Thermosyphons and meltwater feedback have potential to reduce the basal melt rate in areas like Thwaites Glacier.

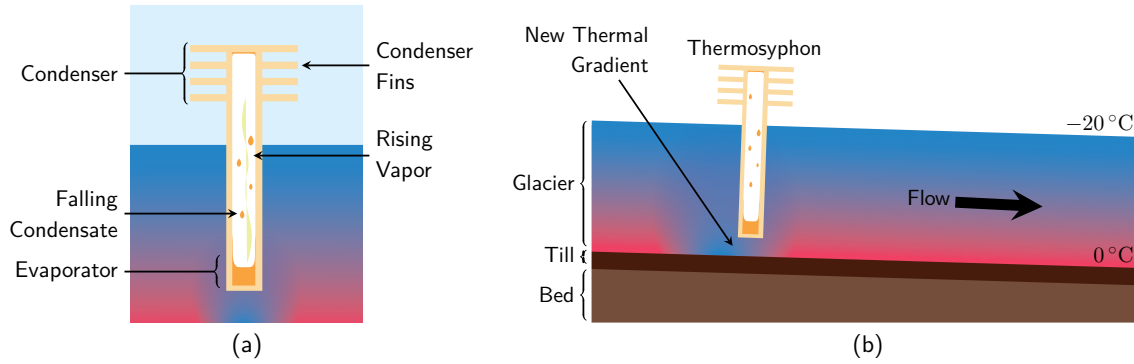


Fig. 4. Diagram of a thermosyphon (a) with parts labeled. (b) in context of an Antarctic ice stream.

1.4. Thermosyphons in Geotechnical Applications

Thermosyphons have been a staple of Arctic geotechnical engineering since the 1960s, to stabilize structures in permafrost (Richardson, 1979; Long, 1963; Wagner, 2014), and have been in use for civil infrastructure in northern USA, northern China (Chen and others, 2021), northern Canada (Zueter and others, 2021) and Russia (Perlshtein, 2008). Notably, thermosyphons have been used to stabilize transmission line towers (Guo and others, 2018), airport civil infrastructure (Jardine and others, 1992; Smith and others, 1991), roadways and railroad tracks (Zarling and Haynes, 1991), and oil pipeline infrastructure (Chen and others, 2021; Heuer, 1979). Thermosyphons have been deployed at massive scale for large civil engineering projects

such as the roughly 120,000 installed for the trans-Alaska pipeline (Heuer, 1979). Reviews of thermosyphon applications in the Arctic have focused on applications (Wagner, 2014), as well as engineering methods (Zarling and Haynes, 1991).

There are many architectures of thermosyphon that may be used for geotechnical applications, including traditional vertical thermosyphons, sloped thermosyphons, flat evaporator thermosyphons, flexible evaporator thermosyphons, and completely buried thermosyphons (Wagner, 2014). These designs may be configured either as a single pipe or as a loop configuration. Additionally, thermosyphons may be passive, or may be “hybrid”, implementing an additional cooling element at the condenser. Thermosyphons configured in a completely vertical orientation are ideal for stabilizing piles and other vertical foundation structures, whereas flat evaporator structures are for stabilizing wide foundations (Wagner, 2014).

Working fluid selection for geotechnical applications varies on the temperature range. For geothermal applications where the device will always be above 0 °C, water is by far the most popular choice due to high specific heat, easy availability and low viscosity. For Arctic applications with temperatures below 0 °C, C₃H₈ (propane), NH₄, and CO₂ are the most commonly used options (Long and Zarling, 2004).

The thermosyphons commonly used for Arctic infrastructure are generally <40 m in length, with most of them substantially shorter (Long and Zarling, 2004; Haynes and others, 1992). For use in an Antarctic ice stream, kilometer-scale devices would be needed, to drive heat from the bed and towards the surface. Long thermosyphons are being tested for geothermal power experiments. For instance, Huang and others (2022) designed, built and tested a 3000 m water-filled thermosyphon which attained 190 kW of sustained heat flux using water as a working fluid. This thermosyphon was constructed out of P110 oil casing tubing which have pre-formed gas-tight seals, and were assembled in-situ using a tower structure (Huang and others, 2022). The tube varied in diameter between 244.5 mm near the surface and 177.8 mm for the bottom half, with wall thicknesses ranging from 11.99 mm to 9.19 mm (Liu and others, 2022; Chen and others, 2022).

1.5. Thermosyphons in Glaciers: Challenges and Opportunities

Compared to traditional deployments in permafrost, use of thermosyphons in an ice stream poses additional challenges and opportunities. The use of thermosyphons in glaciers has previously been proposed (Lockley and others, 2020), but this work is the first to analyze the thermodynamics of the system. As a basal intervention, thermosyphons are of particular interest because they do not require any continuous power input, and the intervention can be quickly aborted if necessary by extracting the internal working fluid. In evaluating the safety and efficacy of thermosyphons for slowing ice streams, there are many physical and human systems factors involved as noted in Figure 1. Many of the technical challenges pertain to the glacier as a whole, while others are specific to thermosyphons. This paper analyzes the thermal performance of thermosyphons in glaciers, then contextualizes how this heat flux would change the flow of an ice stream.

Ice deforms as a non-Newtonian shear-thinning fluid where the viscosity decreases with increased stress (Glen, 1955; Millstein and others, 2022; Ranganathan and Minchew, 2024). Thus, in an ice stream, flow velocity and viscosity depend on depth. Due to the different flow rates around an object placed into a non-linear viscous fluid, such as a thermosyphon in a glacier, there will be shear forces along the length of the object. Since the ice stream is flowing down a slope, the thermosyphon will advect with the

glacier, as shown in Figure 4, but due to the nonlinear vertical shear profile of the glacier, there may be different relative velocities at different points on the thermosyphon. For fast-moving glaciers such as Thwaites Glacier, glacier motion predominantly stems from slip in the subglacial till rather than internal deformation of the ice; thus, the surface velocity is similar to the bed slip velocity. This paper analyzes the thermal performance of thermosyphons without consideration of the mechanics of flow around them; this is valid for glaciers with low vertical shear, an assumption that may be extended using flexible thermosyphon structures or other means. An extended analysis of thermosyphon longevity in glaciers undergoing higher rates of vertical shear will be a topic for future work.

2. THERMAL MODEL

2.1. Thermal Model Description

A critical step in assessing the viability of thermosyphons for glacier remediation is characterizing the transfer of heat from the bed through the thermosyphon towards the surface. The system may be broken down into three parts: convection through the thermosyphon itself, conduction from the bed to the evaporator, and conduction from the condenser to the surface. To facilitate this breakdown, each stage may be characterized as an independent thermal resistance,

$$R = \frac{\Delta T}{Q}, \quad (6)$$

where ΔT is the temperature difference across the stage, and Q is the corresponding heat flow. Most simply, the thermosyphon can be represented as two spherical bodies representing the evaporator and condenser, connected by a finite thermal resistance representing the convective processes inside the thermosyphon. The physical dimensions in this method are represented in Figure 5. In most traditional thermosyphon deployments, the condenser is exposed to the atmosphere to maximize convective heat transfer. However, in potential Antarctic ice stream applications, it is possible that the condenser may become buried by snow during operation; therefore, the thermosyphon is modeled in a buried configuration, since that is a more conservative estimate of heat transfer.

A further important simplification is the assumption that the temperature of the evaporator may be taken to be in steady state, since the timescale of the thermosyphon internal convection is much faster than the timescale of thermal diffusion through the firn and ice. In steady state, where the thermosyphon is much further away from the bed than its effective radius, the thermosyphon evaporator thermal resistance between the evaporator and the bed will be

$$R_{\text{evap,bed}} \approx \frac{1}{2\pi k_{\text{ice}} r_{\text{evap,eff}}}, \quad (7)$$

where k_{ice} is the thermal conductivity of ice, and $r_{\text{evap,eff}}$ is the effective radius of the evaporator. Similarly, between the condenser and the surface, in cases where the condenser is buried, and where the thermosyphon is much further away from the surface than its effective radius, the steady state thermal resistance between

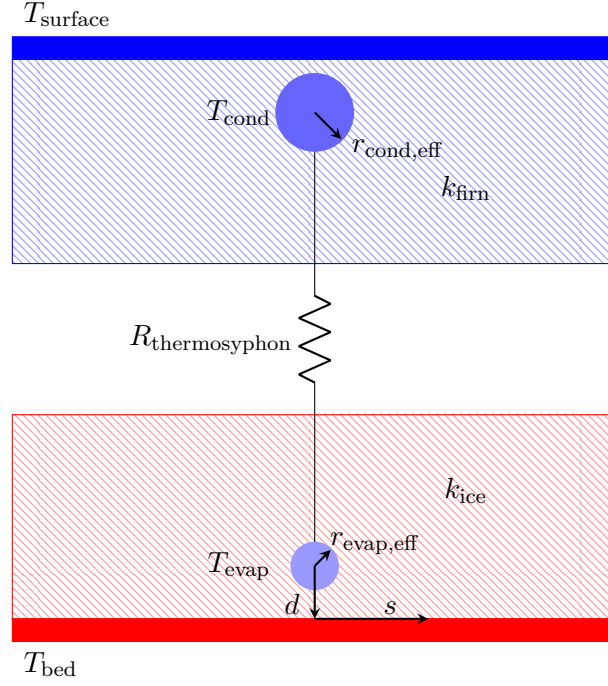


Fig. 5. Thermodynamic schematic of thermosyphon represented by a spherical evaporator and spherical condenser connected with a finite thermal resistor.

the thermosyphon condenser and the surface will be

$$R_{\text{cond,surface}} \approx \frac{1}{2\pi k_{\text{firn}} r_{\text{cond,eff}}}, \quad (8)$$

where k_{firn} is the thermal conductivity of firn near the surface, and $r_{\text{cond,eff}}$ is the effective radius of the condenser. In this paper, $r_{\text{cond,eff}}$ is taken to be 5 m, a value for a large buried fin structure, and a conservative value for thermosyphons exposed to surface wind. Equations 7 and 8 assume that the thermosyphon is much further away from the planar surface than the effective radius of the thermosyphon; this is a conservative assumption, since when the thermosyphon is closer, then the thermal resistance will be significantly lower. For a realistic thermosyphon which is cylindrical rather than spherical, a flux comparison from Laplace's equation can be used to find the equivalent radius of the evaporator

$$r_{\text{evap,eff}} = \frac{L_{\text{evap}}}{2 \left[\ln \left(\frac{2L_{\text{evap}}}{r_{\text{evap,exterior}}} \right) - 1 \right]}. \quad (9)$$

where $r_{\text{evap,exterior}}$ is the radius of the cylindrical evaporator, and L_{evap} is the length of the cylindrical evaporator section (Bergman and Lavine, 2017; Maxwell, 1877). The steady-state temperature of the evaporator, T_{evap} , can be solved for using a thermal resistive divider with the components shown in Figure 5, such that

$$T_{\text{evap}} = (T_{\text{surface}} - T_{\text{bed}}) \frac{R_{\text{evap,bed}}}{R_{\text{cond,surface}} + R_{\text{thermosyphon}} + R_{\text{evap,bed}}}, \quad (10)$$

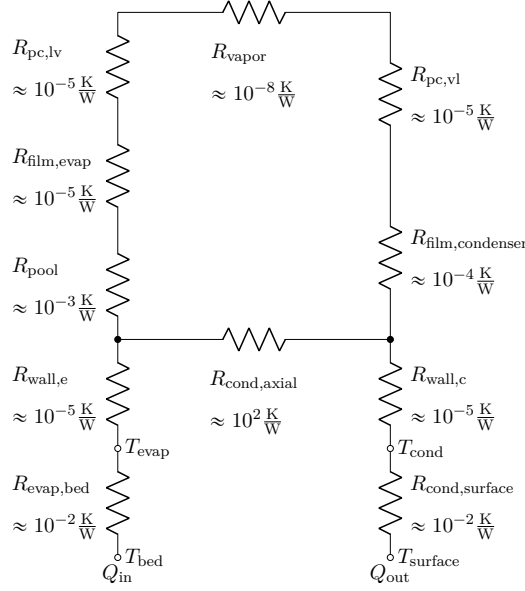


Fig. 6. The convective processes in the thermosyphon are modeled with a set of discrete thermal resistances.

where T_{surface} is the surface temperature, T_{bed} is the bed temperature, and $R_{\text{thermosyphon}}$ is the thermal resistance of the thermosyphon itself. Using this steady-state temperature for T_{evap} , and the melting temperature of the bed at a constant T_{bed} , we can solve for the heat flux driven by the thermosyphon through the plane of the bed.

2.2. Thermosyphon Model

The convective processes in the thermosyphon itself may be broken down into a set of discrete thermal resistance values (Mantelli, 2021). An overview of these processes and calculation methods is presented here, with a full model description provided in the supplementary document. A schematic of this breakdown is shown in Figure 6. The exterior steady-state thermal resistances from the ice and firn, $R_{\text{evap,bed}}$ and $R_{\text{cond,surface}}$, are those in Equation (7) and Equation (8). The next stage is conduction through the walls of the evaporator and condenser stages, which can be evaluated using the dimensions of the evaporator and condenser structures. The conduction through the condenser walls, $R_{\text{wall,c}}$ is calculated similarly.

Heat transfer from the evaporator walls to the working fluid happens in both the evaporator pool and the falling film in the evaporator region; in most cases these two resistances can be modeled in parallel, but as a conservative assumption for long thermosyphons they can be modeled in series, as in Figure 6. In general, the pool in thermosyphons may be in either a natural convection or boiling regime, where boiling results in higher heat-transfer performance; low wall heat fluxes expected in a passive thermosyphon deployment in glaciers mean that careful design considerations in selecting working fluid pressure and evaporator interior topology should be taken in order to provide sufficient heat transfer in the evaporator pool. Since the exterior thermal resistances are high, even sporadic pool boiling is likely sufficient for acceptable thermosyphon performance. Conservatively, the Nusselt number, which describes the ratio of convective to conductive heat transfer can be assumed to be $\text{Nu} = 10$. The heat transfer coefficient for the evaporator film is estimated using a Nusselt-type model benchmarked with experimental data.

The phase change resistances, $R_{pc,lv}$, $R_{pc,vl}$ have been shown to be negligible, on the order of 10^{-5} K W^{-1} , and the vapor path resistance has also been shown to be negligible on the order of 10^{-8} K W^{-1} (Mantelli, 2021). The resistance of the condenser film can be calculated using a laminar film condensation model as a function of the latent heat of vaporization, the physical size of the condenser, the saturation temperature, and the temperature of the condenser wall, calculated using using a resistor divider from the surface boundary (Bergman and Lavine, 2017, 10.31). As a comparison to thermosyphon convective performance, the axial conductivity of a thermosyphon can be calculated. This resistance through axial conduction is several orders of magnitude higher than that of fluid flow, and thus has a negligible effect on performance. Since the thermal resistance of the evaporator film $R_{\text{film,evap}}$ is dependent on the wall heat flux, these equations are evaluated iteratively, with a heat transfer of $Q = \frac{T_{\text{bed}} - T_{\text{surface}}}{R_{\text{cond,surface}} + R_{\text{evap,bed}}}$ used as an initial condition. This simple model was run assuming a CO_2 working fluid, and the parameters in Table 1, yielding resistances of the order of magnitude shown in Figure 6. The thermosyphon in this model is assumed to have a perfectly adiabatic central section between the evaporator and the condenser; in long thermosyphons, that assumption remains a useful first-order approximation, with potential for including wall fluxes in the central section left as future work. The resistance of the thermosyphon itself, excluding ice and firn conduction, represented as $R_{\text{thermosyphon}}$, shown in Figure 5, is on the order of 10^{-3} K W^{-1} . As observed in Arctic permafrost applications (Long and Zarling, 2004), the resistances through the ice and firn are far higher than those of the thermosyphon. More model details for the interior thermal resistance model are given in the supplementary information.

2.3. Resulting Bed Flux

An expression for the heat flux through the glacier bed may be derived by considering thermal conduction through the glacier. In this model, since the bed is modeled with fixed temperature boundary condition as shown in Figure 5, the movement of the glacier due to till deformation does not change the resulting heat flux through the glacier caused by the thermosyphon in the coordinate frame of the thermosyphon. The transient solution for heat flux through the bed is derived using the method of images. The problem is modeled with a sphere representing the evaporator with temperature T_{evap} placed at $z = d$, away from a plane at $z = 0$ which is fixed at T_{bed} , and a second “image” sphere at $z = -d$ with temperature $T_{\text{image}} = -(T_{\text{evap}} - T_{\text{bed}})$; the flux across the bed in this problem is then halved to remove the image sphere. Using this method and a standard solution for conduction around a sphere, the temperature everywhere can be modeled as

$$T(r, t) = T_{\text{bed}} + (T_{\text{evap}} - T_{\text{bed}}) r_{\text{evap}} [\phi(r) - \phi(r')], \quad (11)$$

where $\phi(r) = \frac{1}{r} \text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right)$, $r = \sqrt{x^2 + y^2 + (z - d)^2}$, $r' = \sqrt{x^2 + y^2 + (z + d)^2}$, $\alpha = \frac{k_{\text{ice}}}{\rho_{\text{ice}} c_{p,\text{ice}}}$ is the thermal diffusivity, and t is time. This expression is based on the complementary error function, $\text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right)$, where $\text{erfc}(b) \equiv 1 - \text{erf}(b) = 1 - \frac{2}{\sqrt{\pi}} \int_0^b e^{-t^2} dt$. This temperature solution in Equation (11) is visualized in Figure 7 in the domain where $z > 0$ (excluding the image heat source) with $d = 20 \text{ m}$ from the bed, with $r_{\text{evap,eff}} = 2 \text{ m}$, after 20 years. Halving the flux to account for the image source, a solution for the spatially-dependent,

Table 1. Parameters for thermosyphon model.

Parameter	Value	Description
T_{surface}	$-20\text{ }^{\circ}\text{C}$	Surface temperature boundary
T_{bed}	$0\text{ }^{\circ}\text{C}$	Bed temperature boundary
T_{sat}	$-10\text{ }^{\circ}\text{C}$	Working fluid saturation temperature
r_{external}	0.1 m	Thermosyphon internal radius
r_{internal}	0.09 m	Thermosyphon external radius
$r_{\text{cond,eff}}$	20 m	Thermosyphon condenser effective radius
L_{total}	1000 m	Thermosyphon length
k_{ice}	$2.2\text{ W m}^{-1}\text{ K}^{-1}$	Ice thermal conductivity
k_{firn}	$1.0\text{ W m}^{-1}\text{ K}^{-1}$	Firn thermal conductivity
$c_{\text{p,ice}}$	$2200\text{ J kg}^{-1}\text{ K}^{-1}$	Ice isobaric specific heat
ρ_{ice}	917 kg m^{-3}	Firn density
ρ_l	990 kg m^{-3}	Working fluid liquid density
ρ_v	130 kg m^{-3}	Working fluid vapor density
μ_l	$100 \times 10^{-6}\text{ Pa s}$	Working fluid dynamic viscosity
k_l	$0.115\text{ W m}^{-1}\text{ K}^{-1}$	Working fluid liquid thermal conductivity
h_{fg}	$298 \times 10^3\text{ J kg}^{-1}$	Working fluid latent heat of vaporization
$c_{\text{p,l}}$	$2200\text{ J kg}^{-1}\text{ K}^{-1}$	Working fluid isobaric specific heat
k_{wall}	$237\text{ W m}^{-1}\text{ K}^{-1}$	Thermosyphon wall conductivity
g	9.81 m s^{-2}	Acceleration due to gravity

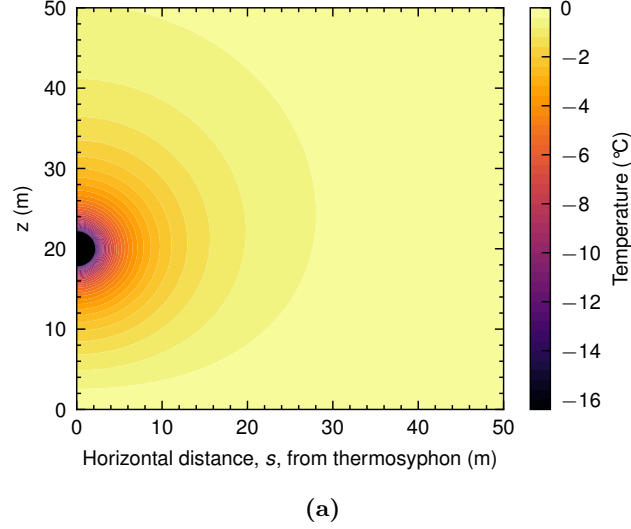


Fig. 7. Temperature around evaporator area of thermosyphon calculated evaporator $d = 20$ m from the bed, with $r_{\text{evap,eff}} = 2$ m, after 20 years.

transient heat flux across the bed due to the thermosyphon may be calculated as

$$q_{\text{bed}}(s, t) = -k \left. \frac{\partial T}{\partial z} \right|_{z=0} = \frac{r_{\text{evap,eff}} dk(T_{\text{evap}} - T_{\text{bed}})}{(d^2 + s^2)^{\frac{3}{2}}} \left[\text{erfc} \left(\frac{\sqrt{d^2 + s^2}}{2\sqrt{\alpha t}} \right) + \frac{\sqrt{d^2 + s^2}}{\sqrt{\pi \alpha t}} \exp \frac{-(d^2 + s^2)}{4\alpha t} \right], \quad (12)$$

where s is the radial coordinate away the origin on the plane of the bed. This solution has a steady state shape of $q_{\text{bed,ss}} = r_{\text{evap,eff}} dk(T_{\text{evap}} - T_{\text{bed}}) (d^2 + s^2)^{(-3/2)}$, with a peak flux of $q_{\text{bed,ss,peak}} = r_{\text{evap,eff}} k(T_{\text{evap}} - T_{\text{bed}})d^{-2}$ and a characteristic time of

$$\mathcal{T} = \frac{d^2}{4\alpha}. \quad (13)$$

The time-varying flux through the bed is visualized in Figure 8a, over time where the center of the evaporator is positioned $d = 20$ m from the bed. This results in a flux distribution that widens over time, and then reaches a steady-state gaussian shape as in space. The steady state characteristics are visualized in Figure 8b, for a variety of values of distance d , showing how further away thermosyphons cause broader spatial flux patterns with a lower peak value. The visualizations in Figure 8 use the parameters in Table 1, and $r_{\text{evap,eff}} = 2$ m. Over 10s of years, assuming a thermosyphon thermal resistance much less than that in the glacier, as found in Section 2.2, the local flux through the bed caused by the thermosyphon is at least the same order of magnitude as the local geothermal heat flux.

In addition to the spatially-varying flux distribution, the total integrated flux through the bed is also useful in designing and understanding thermosyphon spatial arrangements. The flux found in Equation (12)

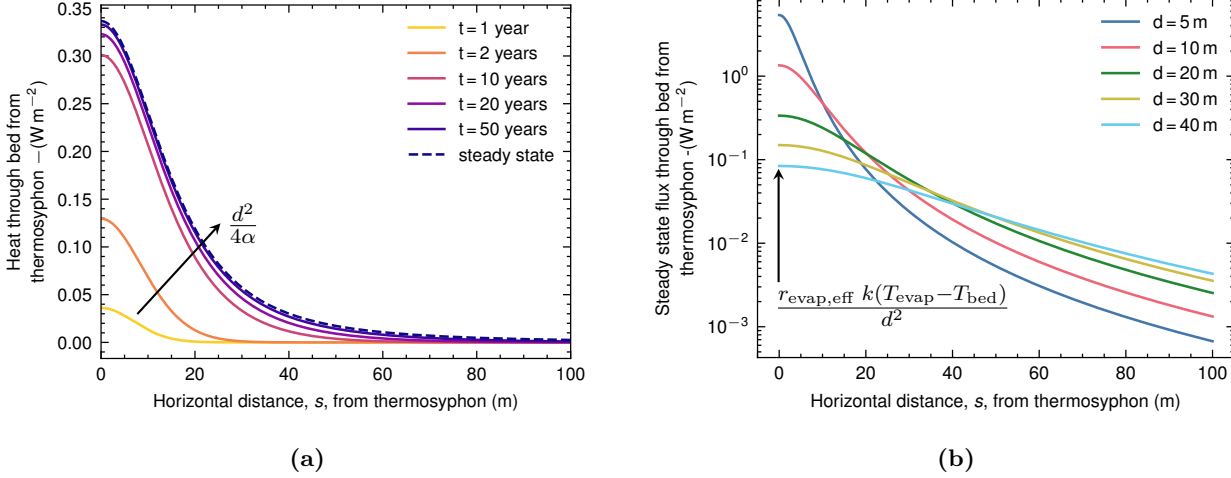


Fig. 8. (a) Visualization of spatially-dependent time-dependent heat flux as a function of spatial radial coordinate s through the bed caused by a thermosyphon with the center of the evaporator positioned 20 m from the bed, with the characteristic time labeled. (b) Visualization of spatially-dependent steady-state heat flux as a function of spatial radial coordinate s with the distance, d , between the thermosyphon and the bed varied. Both plots use the parameters in Table 1, and $r_{\text{evap,eff}} = 2$ m.

is integrated over the plane of the bed, at $z = 0$. The total integrated flux through the bed is

$$\begin{aligned}
 Q_{\text{bed}} &= \int_0^\infty q_{\text{bed}}(s, t) 2\pi s \, ds \\
 &= 2\pi r_{\text{evap,eff}} k (T_{\text{evap}} - T_{\text{bed}}) \operatorname{erfc} \left(\frac{d}{2\sqrt{\alpha t}} \right).
 \end{aligned} \tag{14}$$

The results of this expression are visualized in Figure 9, which compares values of integrated flux as a function of distance from the bed. This expression shows that the total integrated flux across the bed has a steady state which does not depend on d , the distance from the evaporator to the bed, but configurations with larger values of d take longer to reach steady state. Combining the results of Figure 8 and Figure 9, shows that thermosyphons placed at further away distance from the bed, d , have a wider distribution of flux through the bed, but shorter peak flux, and longer time to steady state. In remediation design, these considerations, in addition to the higher mechanical loads near the bed help form design tradeoffs that will inform thermosyphon placement. Since the total system heat flux is bottlenecked by conduction between the bed and the evaporator, and between the condenser and the surface, the size of the evaporator and condenser are critical to the system performance. Equation (14) is evaluated at a range of evaporator sizes, with results in Figure 10 showing this principle. In a glacier installation of thermosyphons, evaporator size would likely be limited by mechanical constraints near the bed, whereas the condenser would not be subject to significant shear loads, and can be made large enough such that $R_{\text{evap,bed}} \gg R_{\text{cond,surface}}$. Thus, in remediation design, the condenser should be made significantly larger than the evaporator, with as much evaporator surface area as design constraints allow.

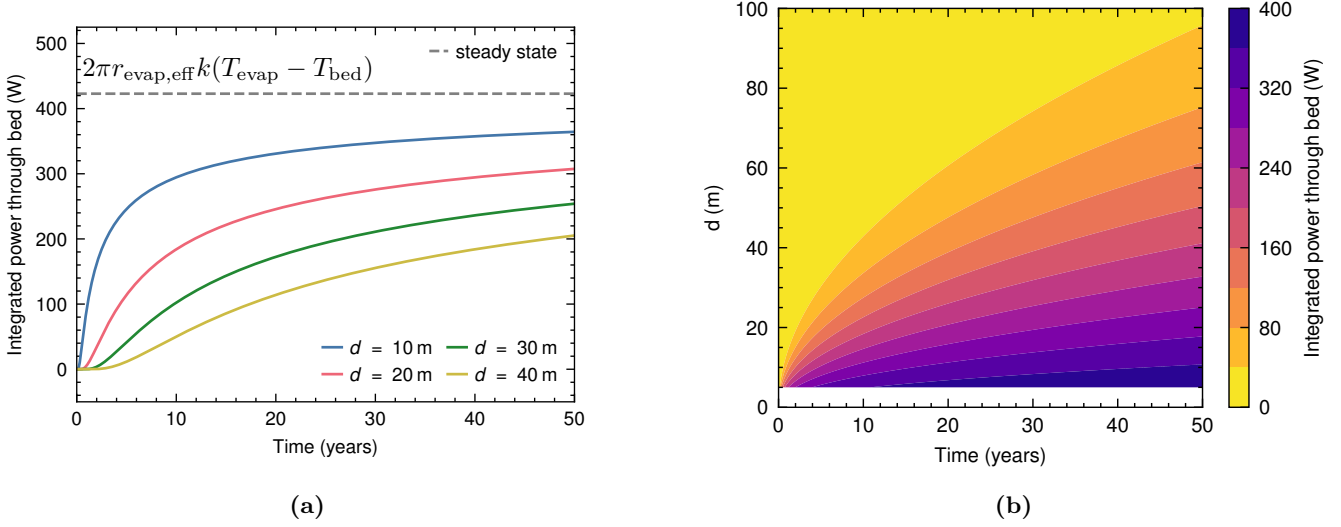


Fig. 9. Integrated total heat flux through the bed of an ice stream, caused by a thermosyphon with an $r_{\text{evap,eff}} = 2$ m with d , the evaporator center distance between the thermosyphon and the bed, varied. Further distances take longer to reach the same steady-state flux. (a) Time series with discrete values of d . (b) Heat map with showing continuous values of d .

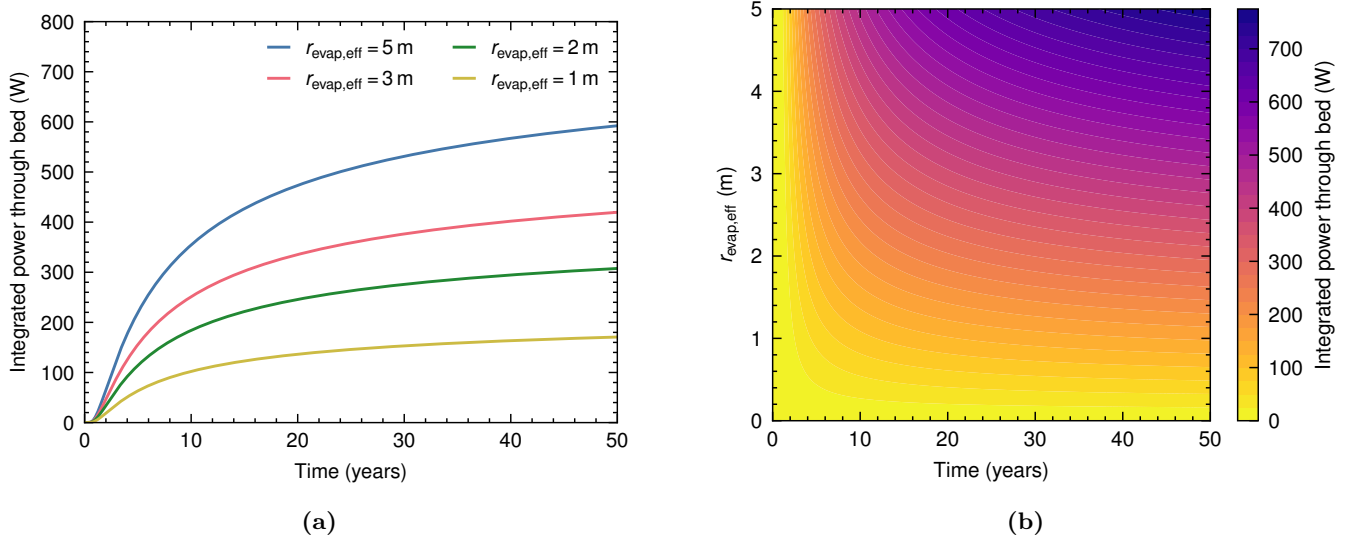


Fig. 10. Integrated total heat flux through the bed of an ice stream caused by a single thermosyphon 20 m from the bed with $R_{\text{evap,eff}}$, the effective radius of the thermosyphon evaporator, varied. (a) Time series with discrete values of $R_{\text{evap,eff}}$. (b) Heat map with showing continuous values of $R_{\text{evap,eff}}$.

3. IMPLICATIONS FOR GLACIER FLOW

When implemented in a glacier, multiple thermosyphons would need to be placed to have a meaningful impact on the melt rate. The integrated flux value from Equation (14) can be multiplied by the number of thermosyphons per area, and converted to melt rate, using Equation (1). For example, after 20 years, and where the evaporator effective spherical radius is set to $r_{\text{evap,eff}} = 2$ m, with the evaporator center placed 20 m from the bed, a basal melt change of -1 mm yr^{-1} may be achieved with 60 thermosyphons per

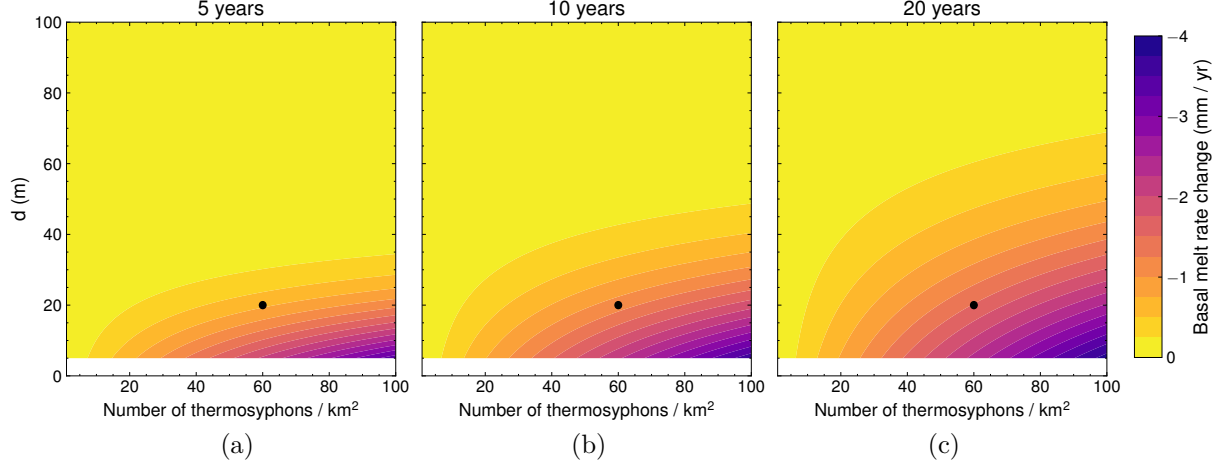


Fig. 11. Spatially-averaged flux value after (a) 5, (b) 10, and (c) 20 years, as a function of the distance, d , from the center of the thermosyphon evaporator to the bed, and the number of thermosyphons, N , where the evaporator effective spherical radius is set to $r_{\text{evap,eff}} = 2$ m. A design point, shown with a black circle at $d = 20$ m with 60 thermosyphons / km^2 is referenced in the text. In many deployment areas, thermosyphons are capable of significantly changing the basal melt rate.

km^2 . Various melt rate perturbations by thermosyphon intervention, $\Delta\dot{m}_{\text{thermosyphon}}$, are calculated and displayed for different distances from the bed and quantities of thermosyphons in Figure 11. Compared to observed melt rates in Figure 3c, the melt rate decrease driven by thermosyphons is of the same order as fluctuations in melt rate caused by variations in geothermal heating and conduction through the glacier. In the context of the ice stream dynamics, the effect of adding thermosyphons acts to nudge the steady state of the ice stream to stagnation as described in Section 1.2, and depicted with an arrow in Figure 2. This model, in Figure 11, shows the static melt perturbation; in the full glacial system, the perturbation from thermosyphons would eventually cause the glacier to slow down, thus further changing the melt rate through indirect feedback.

When applied to Thwaites Glacier, a first-order interpretation of the effect of thermosyphons may be evaluated using a static perturbation of the drag at the bed. We use an initialized Ice Sheet and Sea-level Model (ISSM) (Larour and others, 2012), version 4.24, with a pre-built domain, mesh and drag inversion from Goldberg and others (2026) in this work. The model domain is visualized in the context of ice velocity in Antarctica in Figure 12a. We perturb drag from the initialized state using a parameterization of thermosyphons in Equation (1), with undrained plastic bed model from Tulaczyk and others (2000) to size a static perturbation. This procedure uses a quasi-steady-state approximation (Minchew and others, 2018) to understand the effect of thermosyphons, and the effect of thermodynamic feedback. The model with initialized basal drag (Goldberg and others, 2026) is shown in Figure 12b. When solved to steady-state, this leads to an initial velocity shown in Figure 12c. The initialized basal drag and velocity are used to calculate an initial frictional melt rate as $\dot{m}_{\text{friction}} = \frac{\tau_b U_b}{L_{\text{ice}} \rho_{\text{ice}}}$, shown in Figure 12d.

We selected a spatial patch to add thermosyphons to the system, centered at $(76.2736^\circ \text{S}, 108.6042^\circ \text{W})$ in a fast-flowing region of Thwaites Glacier, shown in Figure 12e. We chose this position because it is far upstream of the grounding zone in an area with relatively rapid flow. A density of 60 thermosyphons per

km^2 over a 10 km radius section of Thwaites is added to achieve $\Delta\dot{m}_{\text{thermosyphon}} = 1 \text{ mm yr}^{-1}$ corresponding to the design point in Figure 11. In this patch, the initial basal drag, τ_0 , is calculated using the Schoof sliding relation included in ISSM (Schoof, 2005),

$$\tau_0 = C_s^2 u_b^m \left[1 + \left(\frac{C_s^2}{C_{\max} N} \right)^{1/m} u_b \right]^{-m}, \quad (15)$$

where C_s is from the initialized ISSM fields, which prescribe $C_{\max} = 0.5$ and $m = \frac{1}{3}$; $N = \rho_i g H - \rho_{\text{water}} g h_{\text{bed}}$ is the effective pressure where ρ_{water} is the density of liquid water and h_{bed} is the depth of the bed below sea level. This effective pressure approximation was chosen to focus the study on the general effect of thermosyphons and thermodynamic feedback without coupling to more complex hydrology. In the undrained plastic bed model,

$$\tau_b = a \exp(-b e), \quad (16)$$

where e is the dimensionless till void ratio, $a = 944 \times 10^6 \text{ kPa}$ and $b = 21.7$ (Tulaczyk and others, 2000). An estimate of the initial till void ratio, e_0 , in the patch is thus $e_0 = -b^{-1} \ln\left(\frac{\tau_0}{a}\right)$ (Tulaczyk and others, 2000). A perturbation to the void ratio due to the reduction in melt rate caused by thermosyphons is calculated as

$$\Delta e = -\frac{\Delta\dot{m}_{\text{thermosyphon}} dt}{Z_{\text{till}}}, \quad (17)$$

where $\dot{m}_{\text{thermosyphon}}$ is the melt rate reduction caused by thermosyphons as previously calculated, $Z_{\text{till}} = 2 \text{ m}$ is the till layer thickness, and $dt = 20 \text{ yr}$ is the timescale of melt rate reduction, which was picked to reflect the timescale of thermosyphon effect and to be longer than transient effects. Using this perturbation in till void ratio, the new perturbed basal shear stress in the area with added thermosyphons is calculated as $\tau_{\text{thermosyphon}} = a \exp[-b(e_0 + \Delta e)]$. The basal drag coefficient C_s is calculated using the perturbed basal stress as

$$C_s = \left[\frac{(\tau_{\text{thermosyphon}} C_{\max} N)^{1/m}}{u_b \left[(C_{\max} N)^{1/m} - \tau_{\text{thermosyphon}}^{1/m} \right]} \right]^{m/2}. \quad (18)$$

This perturbation is added to the ISSM model as a locally higher C_s parameter in basal drag.

The basal drag perturbation from this melt perturbation is calculated using the procedure previously described, leading to the change in drag shown in Figure 12f. In this model, the nonlinearity of the till strengthening process in Equation (16) is critical to the working action of thermosyphons. A parameter to characterize local the strength of quasi-static intervention can be defined as $\alpha = \frac{\tau_{\text{thermosyphon}}}{\tau_0}$; for the placement of thermosyphons in this work, α ranges from 1.005 to 1.11. Based on the static basal drag perturbation, this model shows the velocity reduction after adding thermosyphons in Figure 12g. This velocity reduction represents the static effect of installing thermosyphons in Thwaites Glacier. When the thermodynamic feedback is included, this effect will be further accentuated over time by the processes

described in Section 1.2 and Figure 2. The perturbed frictional melt from the decrease in velocity is shown in Figure 12h.

To illustrate the direct effect of the feedback processes, the frictional melt in Figure 12h is used to calculate a new basal drag perturbation, shown in Figure 12i using the previously described procedure, over a time span of $dt = 20$ yr picked to avoid transient effects. The nonlinear till void ratio relation in Equation (17) drives an asymmetric response to the thermodynamic feedback, accentuating the indirect effect of the thermosyphons. Further melt rate changes from thermosyphons were not included in this feedback step to study the effect of thermodynamic feedback in isolation. When the model is run on this perturbed basal drag field, it leads to a velocity shown in Figure 12j. The reduction in velocity, on the order of 500 m yr^{-1} , due to this feedback is an order-of-magnitude greater than that from the initial perturbation, and illustrates the direction and significance of the thermodynamic feedback processes discussed in Section 1.2. From Figure 12j and Figure 12i, a new melt rate is calculated and shown in Figure 12k. Applying thermodynamic feedback in a quasi-steady state model approximates the magnitude of the feedback mechanism in the full spatial context of Thwaites Glacier, showing the significance of thermodynamic feedback to basal intervention.

4. DISCUSSION

This paper uses a thermal conduction model through the glacier and a lumped model of thermosyphon resistance to identify the effect of thermosyphons and size a perturbation to a flow mechanics model which shows the potential of thermosyphons for use as a glacier remediation strategy. The goal of this work is to provide a first-order model for each part of the physical system, towards understanding the main principles of thermosyphons for glacier remediation. From each modeling step, we find important physical interpretations that guide system understanding. The lumped element interior model of the thermosyphon identifies possible working fluid choices for the glacial environment, with the key finding that in most cases thermosyphons would be expected to have low thermal resistance compared to the surrounding glacial environment. The conduction model in this paper finds that thermosyphons can drive heat flux on the same order as geothermal heat flux, and that the size of the evaporator, $R_{\text{evap,eff}}$, is a performance-limiting design parameter. The glacier flow model in this paper shows that thermosyphons are capable of driving a local velocity reduction that results in significant further thermodynamic feedback-induced slowing; from this initial model, it is clear that understanding the thermodynamic feedback is important for remediation design and planning. Thermosyphons offer a promising method to stabilize Antarctic glaciers and ice streams, such as Thwaites Glacier, adapting existing technology common in the Arctic to potentially reduce sea level rise from ice sheet collapse.

The glacial flow model in this paper uses quasi-steady-state offline-coupled flow mechanics and thermodynamics to illustrate the effect of thermosyphons. Several areas of future work are needed to understand the spatial and temporal effects of thermosyphons. Future 2D and 3D time-dependent flow models like ISSM may be used to determine optimal thermosyphon placement and to build understanding of how sticky spots caused by thermosyphons are related to the simple till void ratio model presented in this paper. The nonlinearity of the till strengthening process in Equation (16) is critical to the working

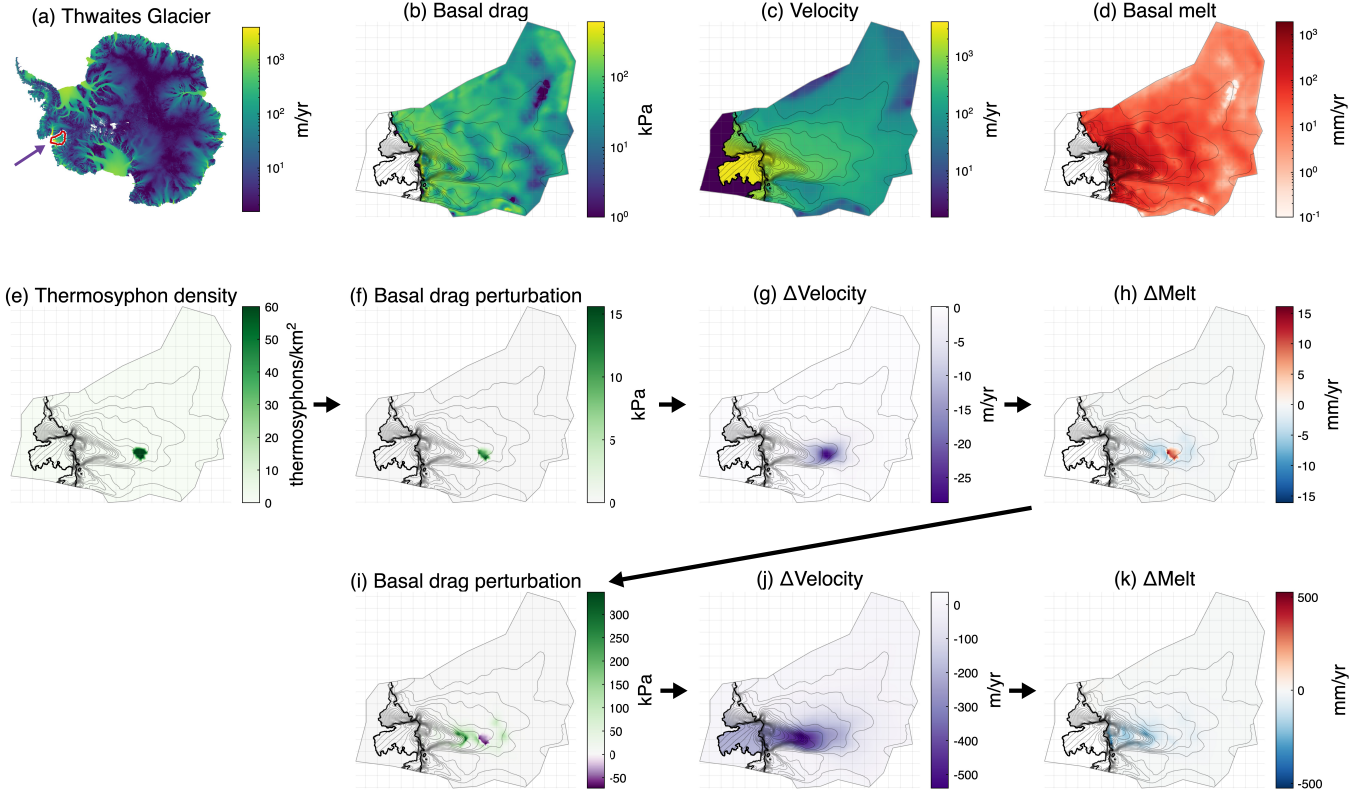


Fig. 12. Model of Thwaites Glacier in Antarctica showing the effect of thermosyphons. (a) shows the ice flow velocity in Antarctica (Rignot and others, 2011, 2017; Mouginot and others, 2012) with the location of Thwaites Glacier noted. (b), (c) and (d) show the initial basal drag field, velocity (with contours every 100 m/yr^{-1}), and frictional melt respectively. (e), (f), (g) and (h) show the added thermosyphon distribution, estimated perturbation to basal drag, and corresponding velocity and frictional melt respectively, representing the static perturbation from adding thermosyphons. (i), (j), and (k) show basal drag perturbation from the change in melt in (h), the subsequent change in velocity, and the new melt rate respectively, illustrating the effect of the thermodynamic feedback that governs basal melt-rates in Antarctic glaciers. The grounding line is shown in black on all plots. All plotted values are within the range of the respective plot colorbars. All plots have a 20 km grid, with the velocity contours from (c) overlaid for spatial reference. Model domain and initialization from Goldberg and others (2026), with grounding line mask from Raspoet and Pattyn (2025).

action of thermosyphons. Collecting and characterizing of additional till samples from Thwaites Glacier can reduce uncertainty around the dynamics modeled in this work. Additionally, the timescale of the slowing processes driven by Equation (17), shown in Figure 12 is highly sensitive to the thickness, Z_{till} , and initial void ratio of the till layer, e_0 . Characterizing the physical properties and initial water present in and near the deforming region is essential for constraining this timescale. Future work could also compare the time-dependent dynamics of various basal interventions, including obstacles, pumping and thermosyphons. More work is needed to understand the mechanical behavior under the vertical shear of ice streams; current results do not model the effect of relative velocity on the heat transfer mechanisms or survivability of thermosyphons. Future models of thermosyphons may also explicitly represent effects such as heat loss through the walls, and be validated against test data from long thermosyphons. If, upon more

modeling and testing, conventional gravity-driven thermosyphons do not provide sufficient heat flux in the configuration used, there are other thermal architectures, such as ultra-low-power single-phase loop devices which are candidates for cooling the bed. Many areas of research into the mechanics and thermodynamics of Antarctic ice streams that are necessary for evaluation of glacier remediation are also critical for general understanding of ice sheet evolution and estimating sea level rise. This overlap ensures that even if basal remediations were shown to be untenable for scientific, engineering or societal reasons, the research involved would still be broadly applicable.

Further work is also needed to engineer particular thermosyphon designs suited for basal interventions, including more detailed modeling of internal convective processes, benchmarking with laboratory experiments, and logistical analysis. Strategies to improve thermosyphon survivability and performance in the glacial environment can also be explored in future work. For instance, flexible or jointed thermosyphons may improve survivability by allowing parts of the thermosyphon to move with vertical shear. Performance of the evaporator may be increased significantly by adding structures that deploy with glacial flow or expand over time, increasing the effective radius of the evaporator, $R_{\text{evap,eff}}$, and significantly reducing the number of thermosyphons needed. Additionally, adding a drilling apparatus to the bottom of the thermosyphon itself may significantly ease deployment logistics.

Evaluating the safety and efficacy of thermosyphons involves integrating analysis of Antarctic ice streams, engineering of thermosyphons themselves and understanding of human systems. In an era where climate change has driven ice-mass loss from the ice sheets, research on basal interventions for remediation is important for understanding the full suite of options for adapting to sea level rise. With the potential option of ice sheet remediation, understanding the stakeholders, policy landscape and relationship to other types of remediation efforts are critical steps. The roadmap in Figure 1 outlines a path towards furthering our understanding of thermosyphon-based remediation efforts, with the goal of bringing policymakers and other stakeholders to informed decisions around basal interventions.

5. CONCLUSIONS

This work investigates the potential of thermosyphons as tools for glacier remediation. The effect of thermosyphons is quantified by modeling the internal thermal resistance of a realistic thermosyphon, then studying an analytical solution for conductive heat flow through the ice stream. Thermosyphons are able to produce a perturbation to melt rate at the bed of a glacier in the same order of magnitude as existing variations in geothermal heat flux, showing potential for their use in glacial interventions. When added to a glacial mechanics model, we illustrate the potential role of thermosyphons in nudging the glacier system towards stagnation through natural feedback. The conduction model in this paper gives insights into the design of thermosyphon to be used in glacial interventions; importantly, the relative size of the evaporator section should be made as large as mechanically feasible to avoid creating a thermal bottleneck. The condenser, especially if buried, should have large fin structures to increase effective radius. Additionally we outline a research roadmap, identifying key areas of focus in glaciology, engineering and human systems in evaluating the safety and efficacy of thermosyphon interventions. Overall, thermosyphons show potential for glacial remediation warranting further research.

6. CODE AVAILABILITY

Code for this project is available at: <https://github.com/MIT-Glacier-Interventions/thermosyphon-thermal-public/>

7. AUTHOR CONTRIBUTIONS

A.M.: conceptualization, analysis, methodology, original draft preparation, funding acquisition. J.C.: analysis, methodology. C.M.: methodology, review and editing, funding acquisition. B.M.: conceptualization, review and editing, funding acquisition.

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9. COMPETING INTERESTS

B.M. and C.M. are co-founders of Arête Glacier Initiative (areteglaciers.org), a non-profit organization that is fiscally sponsored by Digital Harbor Foundation. Arête was founded in 2024 to provide funding to glaciological research community, develop sea-level rise forecasts, and to research glacier remediation. B.M. and C.M.'s consulting work with Arête is separate from their university research activities. A provisional patent application related to specific thermosyphon engineering architectures for glaciers has been filed and assigned to MIT, with A.M., J.C., and B.M. listed as inventors.

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Models of Thermosyphon Performance, Design, and Influence on Glacier Flow, Supplement

Alex S. Miller, Justin T. Cole, Colin R. Meyer, Brent M. Minchew

S1 Derivation of flux through bed with spherical heat source

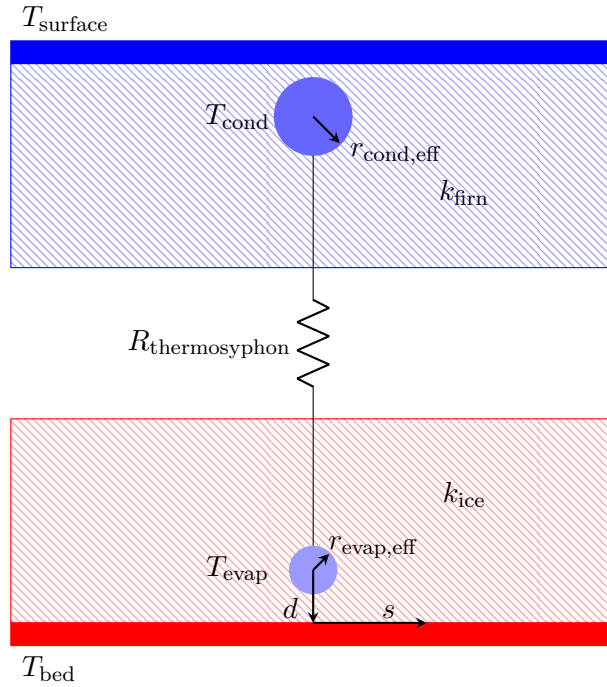


Figure S1: Thermodynamic schematic of thermosyphon represented by a spherical evaporator and spherical condenser connected with a finite thermal resistor.

S1.1 Finding evaporator temperature

A further important simplification is the assumption that the temperature of the evaporator may be taken to be in steady state, since the timescale of the thermosyphon internal convection is much faster than the timescale of thermal diffusion through the firn and ice. In steady state, where the thermosyphon is much further away from the bed than its effective radius ($d \gg r_{\text{evap,eff}}$), the thermosyphon evaporator thermal resistance between the evaporator and the bed will be

$$R_{\text{evap,bed}} = \frac{1}{2\pi k_{\text{ice}} r_{\text{evap,eff}} \left(1 - \frac{r_{\text{evap,eff}}}{2d}\right)} \approx \frac{1}{2\pi k_{\text{ice}} r_{\text{evap,eff}}}, \quad (\text{S1})$$

where k_{ice} is the thermal conductivity of ice, and $r_{\text{evap,eff}}$ is the effective radius of the evaporator [1] (Table 4.1 Case 1). Similarly, between the condenser and the surface, in cases where the condenser is

buried, and where the thermosyphon is much further away from the surface than its effective radius, the steady state thermal resistance between the thermosyphon condenser and the surface will be

$$R_{\text{cond,surface}} = \frac{1}{2\pi k_{\text{firn}} r_{\text{cond,eff}} \left(1 - \frac{r_{\text{cond,eff}}}{2d}\right)} \approx \frac{1}{2\pi k_{\text{firn}} r_{\text{cond,eff}}}, \quad (\text{S2})$$

where k_{firn} is the thermal conductivity of firn near the surface, and $r_{\text{cond,eff}}$ is the effective radius of the condenser [1] (Table 4.1 Case 1). Both Equation (S2) and Equation (S1) assume that the thermosyphon is much further away from the planar surface than the effective radius of the thermosyphon; this is a conservative assumption, since when the thermosyphon is closer, then the thermal resistance will be significantly lower.

The steady-state temperature of the evaporator, T_{evap} , can be solved for using a thermal resistive divider,

$$T_{\text{evap}} = T_{\text{surface}} \frac{R_{\text{evap-to-bed}}}{R_{\text{condenser-to-surface}} + R_{\text{thermosyphon}} + R_{\text{evap-to-bed}}} \quad (\text{S3})$$

where T_{surface} is the surface temperature, T_{bed} is the bed temperature, and $R_{\text{thermosyphon}}$ is the thermal resistance of the thermosyphon itself. Using this steady-state temperature for T_{evap} , and a temperate bed of constant T_{bed} , we can solve for the heat flux caused by the thermosyphon through the plane of the bed.

S1.2 Equivalent shapes

In the derivation of the power through the bed the evaporator is simplified to be a spherical shape with radius $r_{\text{evap,eff}}$ whereas in reality, most thermosyphons are a cylindrical geometry. Thus, we use a shape factor to find the equivalent radius of a given cylindrical thermosyphon. To make a simple comparison, we consider the system in steady state such that it is governed by Laplace's equation. Equivalent shapes have been studied extensively in electrostatics and potential flow which are also governed by Laplace's equation.

Bergman and Levine give equivalent shape factors for a cylindrical body which is attached to a plane of a different temperature, which gives a solution in cases where $L \gg r_{\text{evap,eff}}$, and $d \approx 0$; combining the spherical and cylindrical shape factors listed yields an equivalent radius of

$$r_{\text{evap,eff}} = \frac{L_{\text{evap}}}{2 \ln \left(\frac{2L_{\text{evap}}}{r_{\text{evap,exterior}}} \right)}. \quad (\text{S4})$$

where $r_{\text{evap,exterior}}$ is the radius of the cylindrical evaporator, and L_{evap} is the length of the cylindrical evaporator section [1] (Table 4.1 Case 1 and 3). In cases where the cylinder is moved away from the plane, such as the case with a hypothetical thermosyphon deployment, a small additional correction can be added, which is valid where $L \gg r_{\text{evap,eff}}$, and $d \gg L$, yielding a shape factor correction of

$$r_{\text{evap,eff}} = \frac{L_{\text{evap}}}{2 \left[\ln \left(\frac{2L_{\text{evap}}}{r_{\text{evap,exterior}}} \right) - 1 \right]} \quad (\text{S5})$$

where there is an extra additive term in the denominator [5]. Since the actual configuration is in between these two cases, either would be a reasonable estimate in for $r_{\text{evap,eff}}$. In this work, Equation (S5) is used. In constructing model geometries that use $r_{\text{evap,eff}}$, care must be taken to pick values of d that are realistic relative to the evaporator radius; otherwise spurious results may occur.

S1.3 Integrated power over bed

Here is a derivation for the flux at the bed as a function of time from a spherical source which represents the evaporator of the thermosyphon. First, we do a derivation of the power integrated over the bed since it is simpler than the spatially varying version.

Greens function for temperature around an instantaneous point source or sink of energy E_0 [2] (section 10.2):

$$T(r, t) = \frac{E_0}{\rho c_p (4\pi\alpha t)^{\frac{3}{2}}} \exp\left(-\frac{r^2}{4\alpha t}\right) \quad (\text{S6})$$

where $r = \sqrt{x^2 + y^2 + (z - d)^2}$.

For a continuous point source, emitting or sinking power P :

$$T(r, t) = \int_0^t \frac{P}{\rho c_p (4\pi\alpha t)^{\frac{3}{2}}} \exp\left(-\frac{r^2}{4\alpha t}\right) dt \quad (\text{S7})$$

Calculate heat flux everywhere,

$$\begin{aligned} q_r(r, t) &= -k \nabla T \\ &= -k \frac{\partial T}{\partial r} \\ &= -k \int_0^t \frac{P}{\rho c_p (4\pi\alpha t)^{\frac{3}{2}}} \left(\frac{-2r}{4\alpha t}\right) \exp\left(-\frac{r^2}{4\alpha t}\right) dt \\ &= \frac{Pr}{2(4\pi\alpha)^{\frac{3}{2}}} \int_0^t \frac{1}{t^{\frac{5}{2}}} \exp\left(-\frac{r^2}{4\alpha t}\right) dt \end{aligned} \quad (\text{S8})$$

where $k = \alpha \rho c_p$. For the bed at $z = 0$, at $r = \sqrt{x^2 + y^2 + (z - d)^2}$, we define $s = \sqrt{x^2 + y^2}$. The flux across the bed involves a projection of the of the radial flux q_r onto direction normal to the plane.

$$\begin{aligned} q_z|_{z=0} &= q_r \frac{d}{r} \\ &= \frac{Pd}{2(4\pi\alpha)^{\frac{3}{2}}} \int_0^t \frac{1}{t^{\frac{5}{2}}} \exp\left(-\frac{r^2}{4\alpha t}\right) dt \\ &= \frac{Pd}{2(4\pi\alpha)^{\frac{3}{2}}} \int_0^t \frac{1}{t^{\frac{5}{2}}} \exp\left(-\frac{(s^2 + d^2)}{4\alpha t}\right) dt \end{aligned} \quad (\text{S9})$$

Noting that $\int_0^\infty \exp\left(\frac{-s^2}{4\alpha t}\right) s ds = 2\alpha t$:

$$\begin{aligned} P_{bed} &= \int_0^\infty q_z(r, s, t) 2\pi s ds \\ &= \frac{Pd}{2(4\pi\alpha)^{\frac{3}{2}}} \int_0^t \int_0^\infty \frac{1}{t^{\frac{5}{2}}} \exp\left(\frac{-d^2}{4\alpha t}\right) \exp\left(\frac{-s^2}{4\alpha t}\right) 2\pi s ds dt \\ &= \frac{2\pi Pd}{2(4\pi\alpha)^{\frac{3}{2}}} \int_0^t \frac{1}{t^{\frac{5}{2}}} \exp\left(\frac{-d^2}{4\alpha t}\right) 2\alpha t dt \\ &= \frac{Pd}{4\sqrt{\pi\alpha}} \int_0^t \frac{1}{t^{\frac{3}{2}}} \exp\left(\frac{-d^2}{4\alpha t}\right) dt \end{aligned} \quad (\text{S10})$$

Where $\text{erfc}(t) = 1 - \frac{2}{\sqrt{\pi}} \int_0^t e^{-u^2} du$, do a u -substitution, where: $u = \frac{d}{2\sqrt{\alpha t}}$. Therefore $t = \frac{d^2}{4\alpha u^2}$. This means that $dt = \frac{-d^2}{2\alpha u^3} du$. Additionally $t^{\frac{-3}{2}} = \frac{8\alpha^{\frac{3}{2}} u^3}{d^3}$. Now we change the limits as well, such that $t = 0$ corresponds to $u \rightarrow \infty$ and $t = t$ corresponds to $u = \frac{d}{2\sqrt{\alpha t}}$.

$$\begin{aligned} P_{\text{bed}} &= \frac{P}{2} \text{erfc}\left(\frac{d}{\sqrt{4\alpha t}}\right) \\ &= 2\pi k r_{\text{evap,eff}} (T_{\text{evap}} - T_{\text{bed}}) \text{erfc}\left(\frac{d}{\sqrt{4\alpha t}}\right) \end{aligned} \quad (\text{S11})$$

S1.4 Spatially dependent flux over the bed

Generic temperature from a point source from 1D heat equation [2]:

$$T(r, t) = T_{\text{bed}} + (T_{\text{evap}} - T_{\text{bed}}) \frac{r_{\text{evap,eff}}}{r} \text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) \quad (\text{S12})$$

Now add an image sphere at $z = -d$, noting that $r = \sqrt{x^2 + y^2 + (z - d)^2}$ and $r' = \sqrt{x^2 + y^2 + (z + d)^2}$

$$\begin{aligned} T(r, t) &= T_{\text{bed}} + (T_{\text{evap}} - T_{\text{bed}}) r_{\text{evap,eff}} \left[\frac{1}{r} \text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) - \frac{1}{r'} \text{erfc}\left(\frac{r'}{2\sqrt{\alpha t}}\right) \right] \\ &= T_{\text{bed}} + (T_{\text{evap}} - T_{\text{bed}}) r_{\text{evap,eff}} [\phi(r) - \phi(r')] \end{aligned} \quad (\text{S13})$$

where $\phi(r) = \frac{1}{r} \text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right)$.

$$\begin{aligned} \frac{\partial \phi(r)}{\partial r} &= \frac{\partial}{\partial r} \left[\frac{1}{r} \text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) \right] \\ &= -\frac{1}{r^2} \left[\text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) + \frac{r}{\sqrt{\pi \alpha t}} e^{-\frac{r^2}{4\alpha t}} \right] \\ &= -\frac{\psi(r)}{r^2} \end{aligned} \quad (\text{S14})$$

where $\psi(r) \equiv \left[\text{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) + \frac{r}{\sqrt{\pi \alpha t}} e^{-\frac{r^2}{4\alpha t}} \right]$. On the plane where $z = 0$, $\rho \equiv \sqrt{d^2 + s^2} = r = r'$, and $s = \sqrt{x^2 + y^2}$, $\frac{\partial r}{\partial z} = -\frac{d}{\rho} \Big|_{z=0}$ and $\frac{\partial r'}{\partial z} = \frac{d}{\rho}$.

$$\frac{\partial T}{\partial z} \Big|_{z=0} = (T_{\text{evap}} - T_{\text{bed}}) r_{\text{evap,eff}} \frac{\partial \phi}{\partial \rho} \left(\frac{-d}{\rho} - \frac{d}{\rho} \right) = (T_{\text{evap}} - T_{\text{bed}}) r_{\text{evap,eff}} \left(-\frac{\psi}{\rho^2} \right) \left(\frac{-2d}{\rho} \right) \quad (\text{S15})$$

Halving the contribution from the image source, we evaluate the flux at the bed. Generally, this would be negative since flux towards the evaporator by convention, but we evaluate it as positive for convenience.

$$\begin{aligned} q_{\text{bed}}(s, t) &= k \frac{\partial T}{\partial z} \Big|_{z=0} \\ &= \frac{r_{\text{evap,eff}} dk (T_{\text{evap}} - T_{\text{bed}})}{\rho^3} \psi(\rho) \\ &= \frac{r_{\text{evap,eff}} dk (T_{\text{evap}} - T_{\text{bed}})}{(d^2 + s^2)^{\frac{3}{2}}} \left[\text{erfc}\left(\frac{\sqrt{d^2 + s^2}}{2\sqrt{\alpha t}}\right) + \frac{\sqrt{d^2 + s^2}}{\sqrt{\pi \alpha t}} \exp\left(-\frac{(d^2 + s^2)}{4\alpha t}\right) \right] \end{aligned} \quad (\text{S16})$$

In addition to Figure 8b in the body of the paper, more cases for evaluation of this equation for realistic thermosyphon parameters listed in the body of the paper are shown in Figure S2. Now, we integrate over the plane of the bed, at $z = 0$. We do a change of variables, noting that when $s = 0$, $\rho = d$. Note that, by definition,

$$\begin{aligned}
P_{\text{bed}} &= \int_0^\infty q_{\text{bed}}(s, t) 2\pi s ds \\
&= \int_d^\infty q_{\text{bed}}(\rho, t) 2\pi \rho d\rho \\
&= 2\pi r_{\text{evap,eff}} dk(T_{\text{evap}} - T_{\text{bed}}) \int_d^\infty \frac{\psi(\rho)}{\rho^2} d\rho \\
&= 2\pi r_{\text{evap,eff}} dk(T_{\text{evap}} - T_{\text{bed}}) \left[-\frac{1}{\rho} \operatorname{erfc}\left(\frac{r}{2\sqrt{\alpha t}}\right) \right] \Big|_d^\infty \\
&= 2\pi r_{\text{evap,eff}} dk(T_{\text{evap}} - T_{\text{bed}}) \frac{1}{d} \operatorname{erfc}\left(\frac{d}{2\sqrt{\alpha t}}\right) \\
&= 2\pi r_{\text{evap,eff}} k(T_{\text{evap}} - T_{\text{bed}}) \operatorname{erfc}\left(\frac{d}{2\sqrt{\alpha t}}\right)
\end{aligned} \tag{S17}$$

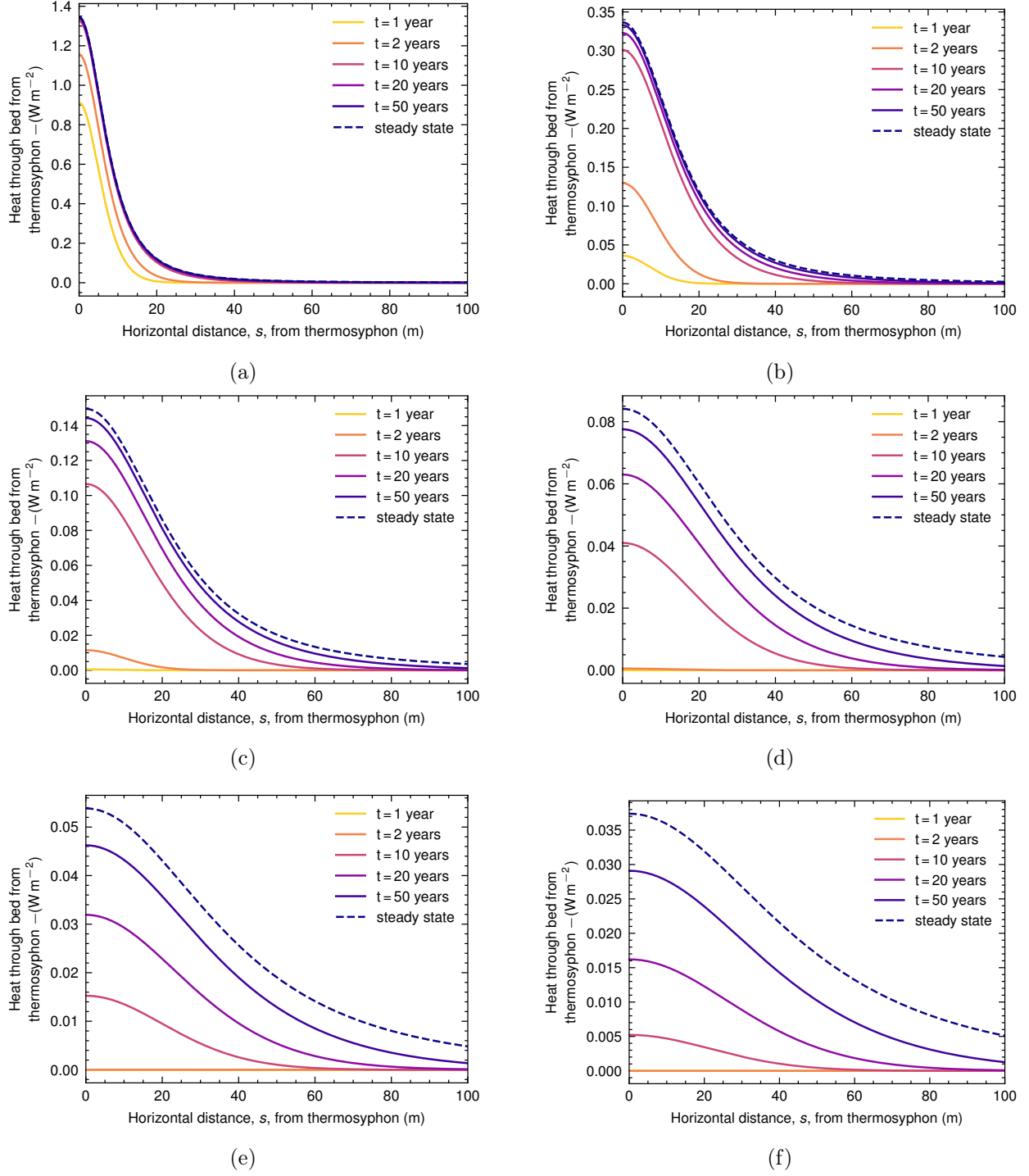


Figure S2: Visualization of time-dependent flux as a function of spatial radial coordinate s through the bed caused by a spherical thermosyphon with parameters in Table 1 in the main paper with the center of the evaporator positioned (a) 10 m from the bed, (b) 20 m from the bed, (c) 30 m from the bed, (d) 40 m from the bed, (d) 50 m from the bed, (d) 60 m from the bed. Note that vertical axes differ between plots within this figure.

S2 Thermosyphon Interior Model

S2.1 Thermosyphon Model

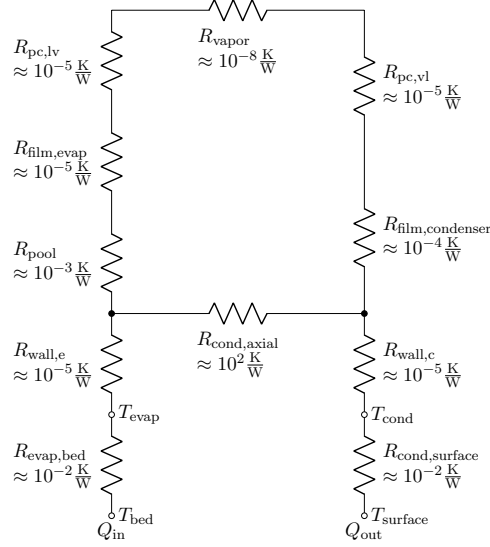


Figure S3: Thermal resistance diagram modeling the interior of the thermosyphon.

The convective processes in the thermosyphon itself may be broken down into a set of discrete thermal resistance values. A schematic of this breakdown is shown in Figure S3. The exterior steady-state thermal resistance values are those in Equation (S1) and Equation (S2). The next stage is conduction through the walls of the evaporator and condenser stages, which can be evaluated as

$$R_{\text{wall,e}} = \frac{\ln \left(\frac{r_{\text{evap,exterior}}}{r_{\text{evap,interior}}} \right)}{2\pi k_{\text{wall}} l_{\text{evap}}}, \quad (\text{S18})$$

where $r_{\text{evap,exterior}}$ is the exterior radius in the evaporator, $r_{\text{evap,interior}}$ is the interior radius of the evaporator, l_{evap} is the length of the evaporator, and k_{wall} is the thermal conductivity of the wall material, as in [4] (4.4). The conduction through the condenser walls, $R_{\text{wall,c}}$ is calculated similarly. Heat transfer from the evaporator walls to the working fluid happens in both the evaporator pool, and the falling film in the evaporator region; these two resistances can be modeled in parallel, but for long thermosyphons it is more conservative to model them in series as in Figure S3. In general, the pool in thermosyphons may be in either a natural convection or nucleate boiling regime; the relatively low wall heat fluxes expected in a passive thermosyphon deployment in glaciers mean that careful engineering needs to be done to make sure that the pool is in at least intermittent boiling. The Nusselt number, which describes the ratio of convective to conductive heat transfer can be assumed to be $\text{Nu} = 10$. The pool resistance can be calculated as

$$R_{\text{pool}} = \frac{r_{\text{evap,interior}}}{\text{Nu} k_l A_{\text{pool}}}, \quad (\text{S19})$$

where $A_{\text{pool}} = 2\pi r_{\text{evap,interior}} L_{\text{pool}}$. This resistance is in parallel with the resistance of the evaporator film,

$$R_{\text{film}} = \frac{1}{h_{\text{film}} A_{\text{film}}}, \quad (\text{S20})$$

where h_{film} is a heat transfer coefficient and $A_{\text{film}} = 2\pi r_{\text{evap,interior}} L_{\text{film}}$ [4] (4.2). The heat transfer coefficient for the evaporator film is estimated using a Nusselt-type model benchmarked with experimental data as

$$h_{\text{film}} = 0.925 k_l \left[\frac{\rho_l(\rho_l - \rho_v)g}{\mu_l^2 \text{Re}_{\text{film}}} \right]^{\frac{1}{3}}, \quad (\text{S21})$$

where k_l is the thermal conductivity of the liquid, ρ_l is the density of the liquid, ρ_v is the density of the vapor, g is the acceleration of gravity, μ_l is the dynamic viscosity of the liquid, and Re_{film} is the Reynolds number of the film layer [4] (4.18). This Reynolds number can be calculated as

$$\text{Re}_{\text{evap,film}} = 2 \frac{q_{\text{evap}}}{\pi r_{\text{evap,interior}} \mu_l h_{lv}}, \quad (\text{S22})$$

where q_{evap} is the heat flux through the evaporator wall, and h_{lv} is the latent heat of vaporization. In the case of a thermosyphon where the pool is in the natural convection regime, $R_{\text{evap,film}} \ll R_{\text{pool}}$, meaning the characteristics of the film region will dominate. The phase change resistances, $R_{\text{pc,lv}}$, $R_{\text{pc,vl}}$ been shown to be negligible, on the order of 10^{-5} K W^{-1} , and the vapor path resistance has also been shown to be negligible on the order of 10^{-8} K W^{-1} [4]. The resistance of the condenser film can be calculated as

$$R_{\text{film,condenser}} = \frac{1}{h_{\text{film,condenser}} A_{\text{pool}}} \quad (\text{S23})$$

where $h_{\text{film,condenser}}$ is a heat transfer coefficient, and $A_{\text{cond}} = 2\pi r_{\text{cond,interior}} L_{\text{cond}}$. This heat transfer coefficient can be calculated using a laminar film condensation model as

$$h_{\text{film,condenser}} = 0.943 \left[\frac{g \rho_l (\rho_l - \rho_v) h'_{fg} L_{\text{cond}}^3}{\mu_l k_l (T_{\text{sat}} - T_{\text{cond,wall}})} \right]^{\frac{1}{4}} \quad (\text{S24})$$

where h'_{fg} is a corrected latent heat of vaporization, L is the length of the condenser portion, T_{sat} is the saturation temperature, and $T_{\text{cond,wall}}$ is the temperature of the condenser wall, calculated using using a resistor divider from the surface boundary [1] (10.31). The corrected latent heat of vaporization is expressed as

$$h'_{fg} = h_{fg} + 0.68 c_{p,l} (T_{\text{sat}} - T_{\text{cond,wall}}) \quad (\text{S25})$$

where h'_{fg} is the latent heat of vaporization, and $c_{p,l}$ is the specific heat of the liquid at constant pressure [1] (10.27). As a comparison, the axial conductivity of a thermosyphon can be calculated as well as

$$R_{\text{cond,axial}} = \frac{l_{\text{eff}}}{\pi k_{\text{wall}} (r_{\text{external}}^2 - r_{\text{internal}}^2)}, \quad (\text{S26})$$

where l_{eff} is the effective length of the thermosyphon, spanning from half of the length of the evaporator to half the length of the condenser. This resistance through axial conduction is several orders of magnitude higher than that of fluid flow, and thus has a negligible effect on performance. Since Equation (S22) is dependent on the wall heat flux, these equations are evaluated iteratively, with a heat transfer of $Q = \frac{T_{\text{bed}} - T_{\text{surface}}}{R_{\text{cond,surface}} + R_{\text{evap,bed}}}$ used as an initial condition. This simple model was run assuming a CO_2 working fluid, and the parameters in Table 1, yielding resistances of the order of magnitude shown in Figure S3. As observed in arctic permafrost applications [3], the resistances through the ice and firn are far higher than those of the thermosyphon. The resistance of the

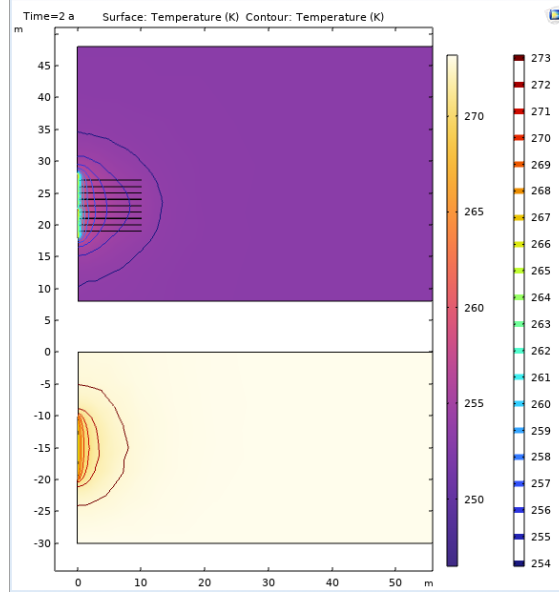


Figure S4: A finite element model of the conduction processes around a thermosyphon in an axisymmetric glacier.

thermosyphon itself, excluding ice and firn conduction is represented as $R_{\text{thermosyphon}}$, shown in Figure S1, is on the order of 10^{-3} K W^{-1} .

S2.2 Model Validation

S2.2.1 Finite Element System Thermal Model

To compare the results found in the continuum thermal model in Section 2.3 with a more realistic geometry, a finite element model was built in COMSOL. This model includes a buried fin structure on the condenser to correspond with the schematic in Figure S1, and consists of a section of ice around the evaporator, a section of firn around the condenser, and a constant lumped thermal resistance connecting them, $R_{\text{thermosyphon}}$ found using the procedure in Section S2.1. This model was constructed assuming that the thermosyphon would be stationary in the ice stream; that is, the rate of convection exceeding the rate of flow. With this assumption, the system is modeled using an axisymmetric 2D coordinate system, where the temperature at the surface and the bed are set as constants. The finite element model shows the temperature change throughout the ice near the bed, and the total heat flux through the bed. Since the model does not account for the latent heat of liquid water near the bed, the total heat flux is a more useful comparison metric than the temperature profile. An image of the COMSOL model setup is shown in Figure S4. If the condenser is buried, the surface firn will have significantly lower thermal conductivity compared to the ice near the bed, as a consequence, large buried fin arrays may be used on the condenser. In COMSOL, these are modeled as a set of metal plates that extend into the ice, spaced evenly on the condenser surface; the outer radius of this fin structure is set to the effective radius of the condenser in Equation (S2).

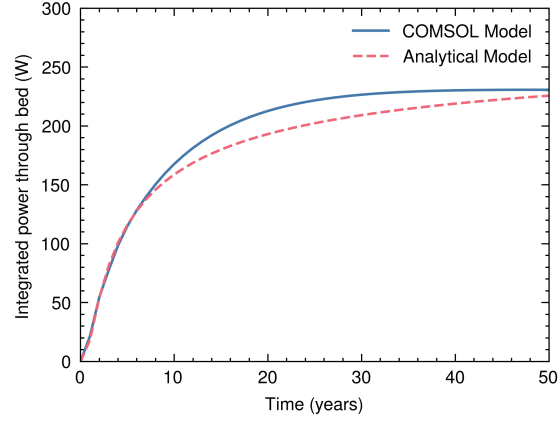


Figure S5: Comparison between analytical approach developed in the main paper Section 2.3 and finite element model.

S3 Basal Perturbation Model

The Figure S6 shows an extended version of Figure 12 which includes more detailed values of the perturbation parameters used.

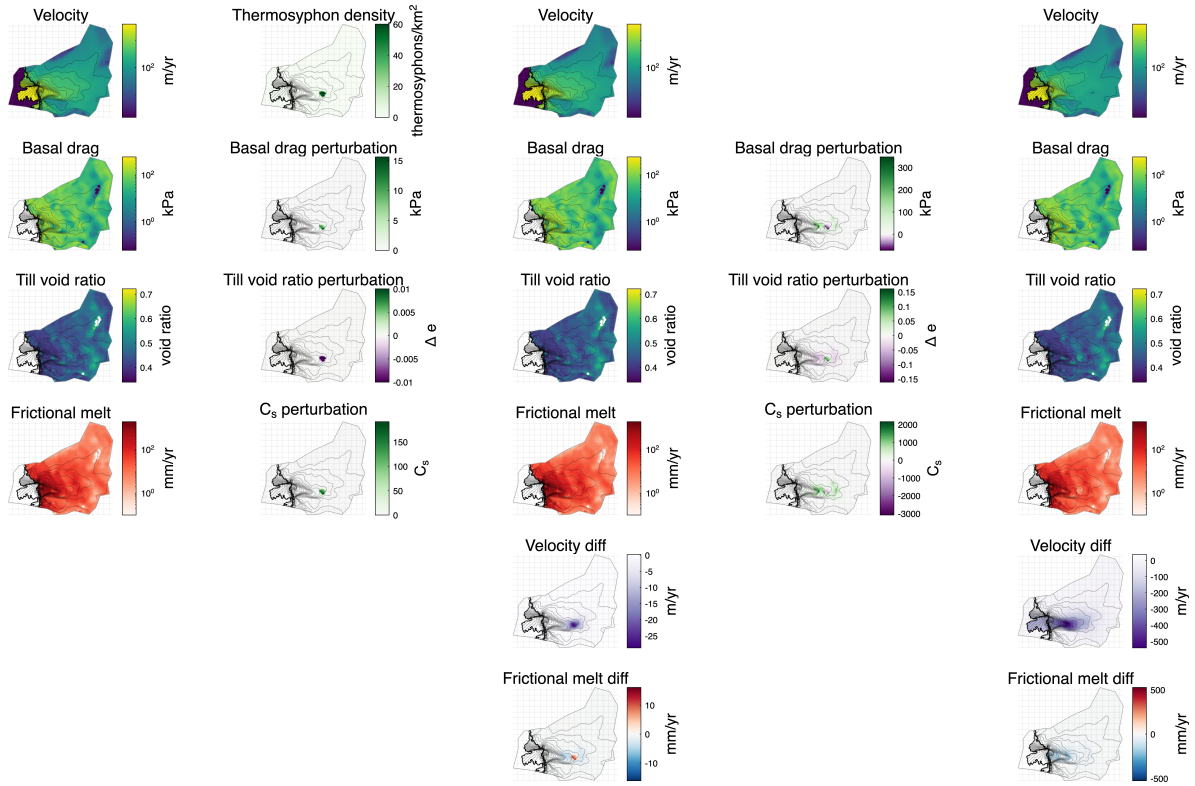


Figure S6: Extended version of Figure 12 which includes the initial state of Thwaites glacier (first column), the calculated perturbation to basal drag due to thermosyphons (second column), the resulting new state of Thwaites glacier (third column), a further meltwater feedback perturbation (fourth column), and the resulting new state of Thwaites glacier (fifth column).

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