A Silicon Microturbopump for a Rankine-Cycle Power Generation Microsystem—Part I: Component and System Design

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Abstract—This paper presents the design approach for a microturbopump, which is the core component of a micro steam turbine power plant-on-a-chip that implements the Rankine thermodynamic cycle for micro power generation. The turbopump integrates components that are demonstrated for the first time at microscale, such as a four-stage radial planar type microturbine and a one-sided hydrostatic thrust bearing (TB) system, along with a spiral groove viscous pump, a partially grooved seal, and a hydrostatic journal bearing. This paper presents the analytical models developed for each component, including a flow resistance model for the TB and models based on lubrication theory for the pump and seal. They are integrated to enable the microsystem design by satisfying force and power balance conditions on the rotor. Considering our previous thermodynamic cycle analysis on the Rankine micro power generation system, which is aimed at generating a few watts of electric power for applications in portable electronics or waste energy harvesting, we have designed a centimeter-scale demo turbopump device delivering 4.7 W of turbine mechanical power and 71% of turbopump efficiency in order to demonstrate the effectiveness of the component design models and system design principles. Fabrication and testing of the microturbopump are presented in the second part of this two-part paper. [2010-0075]

Index Terms—Heat engines, microfluidics, micropumps, power generation, turbines.

NOMENCLATURE

General definition of symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>(\dot{m})</td>
<td>Mass flow rate.</td>
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<tr>
<td>(P)</td>
<td>Pressure.</td>
</tr>
<tr>
<td>(\Delta P)</td>
<td>Pressure difference.</td>
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<tr>
<td>(T)</td>
<td>Torque.</td>
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<tr>
<td>(W)</td>
<td>Power.</td>
</tr>
<tr>
<td>(\mu)</td>
<td>Viscosity.</td>
</tr>
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\(\omega\) | Angular velocity. |
\(\rho\) | Density. |

**Turbine**

\(P_{01}\) | Blade inlet total pressure. |
\(P_{02}\) | Blade outlet total pressure. |
\(P_2\) | Blade outlet static pressure. |
\(K\) | Blockage. |
\(\Delta r\) | Radial position from the flow inlet. |

**Pump**

\(a_1\) | Groove width. |
\(a_2\) | Ridge width. |
\(A\) | \(H^2(1 + \cot^2 \alpha)(\gamma + H^3)/(1 + \gamma H^3)(\gamma + H^3) + H^3 \cot^2 \alpha(1 + \gamma)^2)\). |
\(A_1\) | \((1/K_1)[(-\gamma(1 - H)(1 + \gamma H^3) + S \gamma(\cot \alpha)/(1 - H^3)/(1 + \gamma H^3)(\gamma + H^3) + H^3(\cot^2 \alpha)/(1 + \gamma)^2)]\). |
\(A_2\) | \((1/K_2)[(-\gamma(1 - H)(1 + \gamma H^3) + S \gamma(\cot \alpha)/(1 - H^3)/(1 + \gamma H^3)(\gamma + H^3) + H^3(\cot^2 \alpha)/(1 + \gamma)^2)]\). |
\(B\) | \((3 \gamma H \cot \alpha(1 - H)(1 - H^3)/(1 + \gamma H^3)(\gamma + H^3) + H^3 \cot^2 \alpha(1 + \gamma)^2)\). |
\(C_2^*\) | Leakage coefficient, \(C_2^*(\alpha, H, \gamma, \lambda, k) = \left(e^{-(\pi/6)(1/(\alpha/90))}\tan \alpha/2/(1 + \gamma)(1-H^3)/(1+H^3)\right) Fr_2 - \lambda^2 \varepsilon H \gamma(\pi/k)(1/(\alpha/90))\tan \alpha/2/(1 + \gamma)(1-H^3)/(1+H^3) Fr_1 / (1 - \lambda^2)^3\). |
\(Fr_1\) | Correction factor for the end effect, \((A_1 \cot \alpha/A_1 \cot \alpha + C_1)\). |
\(Fr_2\) | Correction factor for the end effect, \((A_2 \cot \alpha/A_2 \cot \alpha + C_2)\). |
\(g_1\) | \((\gamma H^2 \cot \alpha(1 - H)(1 - H)^3)/(1 + \gamma H^3)(\gamma + H^3) + H^3 \cot^2 \alpha(1 + \gamma)^2)\). |
\(g_2\) | \((g_2^2(\alpha, H, \gamma)/1 + \gamma)\). |
\(g_2^*\) | \((\gamma + H) + (3 \gamma H(1 - H)^2(1 + \gamma H^3)/H^2(1 + \gamma)(1 + \gamma H^3) + H^3 \cot^2 \alpha(1 + \gamma)^2)\). |
\(h_1\) | Gap between groove and the rotor. |
\(h_2\) | Gap between ridge and the rotor. |
\(H\) | \(h_2/h_1\). |
\(K_1\) | \(-(h_2^4 P_0/6 \mu \omega r_1 \Delta r)\). |
\(K_2\) | \(-(h_2^4 P_0/6 \mu \omega r_2 \Delta r)\). |
\(P_0\) | Ambient pressure. |
\(r_1\) | Inner radius. |
\(r_2\) | Outer radius. |
\(\Delta r\) | \(r_2 - r_1\). |
Many kinds of microscale heat engines have been proposed, but the small-size technology and low unit cost from the batch fabrication of micro components bring the benefits of high power production per unit volume from micro heat engines. This provides an alternative to batteries, providing a good candidate for micro power generation, leveraging the technology initially developed for micro gas turbines. The Rankine cycle is similar to the Brayton cycle of gas turbine engines in that they use a turbine to convert fluid energy to mechanical energy. However, the fundamental difference lies in the pressurization process: The Rankine cycle uses a liquid pump instead of a gas compressor. This reduces the compression work significantly because pumping a liquid requires less power than compressing a gas. This makes the Rankine cycle more amenable to miniaturization due to the minimal overall impact of viscous losses on the compression process at small scale than the Brayton cycle, whose efficiency is significantly reduced at lower Reynolds numbers (small scale) [2]. This illustrates that excessive losses can negate the benefit of high power-density production when miniaturizing heat engines. To evaluate the potential system-level performance of a Rankine microengine, a design study is therefore essential, accounting for the limitations of the microfabrication approach and the component behaviors at small scale.

This paper provides the design basis for the development of a microturbopump device as part of a steam turbine power plant-on-a-chip, which implements the Rankine thermodynamic cycle for micro power generation using MEMS technology. The turbopump consists of a multistage turbine driving a spiral groove viscous pump with a partially grooved seal, a hydrostatic thrust bearing (TB) system. Among the components, the turbine and TB have novel configurations that are different from that of the previous MEMS gas turbomachinery [8]. The turbine has a multistage configuration to potentially achieve high pressure ratio (> 10) required for reasonable efficiencies (1%–11%) of the whole micro Rankine thermal system to compete with conventional power sources such as batteries [4], and the TB system is designed to exert the thrust force only from one side with the goal of developing a self-supporting bearing system. The pump and seal have spiral groove patterns on the surface to draw and pressurize the fluids through viscous forces. In addition, dynamic pressurization of the seal was necessary through the use of spiral patterns to confine the liquid flow in a specific area.

This paper begins with a brief overview of a Rankine microturbine power plant-on-a-chip to provide a framework for this study. Then, the focus will move on to the rotating subsystem, referred to as the microturbopump. In the subsequent sections, the function and modeling of each component will be described. Preliminary designs of the components will also be presented to evaluate the design space and achievable performance. The

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**I. Introduction**

As high-tech products such as portable electronics, distributed sensors, and small-scale actuators evolve, the demand for compact power sources is increasing. Although battery technology is still developing, chemical batteries have inherent limits in energy density compared to hydrocarbon fuels. In this context, microelectromechanical systems (MEMS) heat engines can offer an alternative to batteries, providing the benefits of high power production per unit volume from the small-size technology and low unit cost from the batch microfabrication technology.

Since the term “power MEMS” was first suggested to describe microsystems which generate power or pump heat [1], many kinds of microscale heat engines have been proposed.
design approach for the whole system will then be presented along with a specific design for a microturbopump device that has been fabricated for demonstration of the concept and that will be presented in the second part of this two-part paper. The final design of the device has been defined considering various factors such as optimal operation of each component, overall configuration of the device, and the fabrication process to achieve a reasonable performance in view of the Rankine-cycle analysis [4]. Therefore, the final designs of some components are far different from the preliminary designs. Portions of the device design and its demonstration were initially presented in [9].

II. OVERVIEW OF THE RANKINE MICROTURBINE STEAM POWER PLANT-ON-A-CHIP

A typical Rankine vapor power cycle consists of pumping, boiling, expansion, and condensation processes. As shown in Fig. 1, a working fluid in liquid phase is compressed with a pump (states 3 to 4) and then evaporated and superheated to a maximum cycle temperature (state 1, \( T_{\text{max}} \)) through an evaporator. The working fluid in vapor form then expands through a turbine (states 1 to 2) to provide mechanical shaft power to drive the pump and an integrated generator. The cycle is closed by condensing the working fluid (states 2 to 3) by rejecting heat to the surroundings through a condenser. Among the four processes, pumping and expansion happen in the microturbopump device through a silicon microfabricated pump and turbine, respectively.

The Rankine microturbine power plant-on-a-chip concept [10] implements this closed Rankine power cycle using a high-speed microturbine with an integrated pump and generator, as well as on-chip or off-chip heat exchangers. Based on our thermodynamic cycle analysis, each 1-cm² power plant chip is expected to generate between 1 and 10 W of electrical power [10]. A heat supply, heat sink, and power electronics are also required to form a complete power generation system. The heat can come from a miniature combustor, solar radiation, the exhaust of an engine, or other sources of waste heat.

Most of the parts of this device can be implemented using Si bulk micromachining techniques such as photolithography, deep reactive-ion etching, and wafer-bonding techniques, which enables forming quasi 3-D structures of planar extruded shapes [8].

III. MICROTURBOPUMP

The rotating subsystem of the Rankine microturbine power plant-on-a-chip converts the thermal fluidic energy to mechanical energy. The microturbopump, without the electric power generator, will be the focus of this paper.

The microturbopump consists of a turbine, pump, main TB, auxiliary TB, JB, and seals as shown in Fig. 2. Structurally, it has one moving part, static structures, and flow channels. The disk-shaped rotor is enclosed in the static structure and has turbine blades on its top side and the pump on the opposite side. Space is kept for a future generator outboard of the pump.

As shown by the flow paths in Fig. 2, pressurized gas enters from the top side and flows radially outward through the blade rows of the turbine, spinning the rotor (“turbine flow”). Simultaneously, water is drawn through the pump on the bottom side (“pump flow”). Liquid flow can either be guided to the turbine side through holes in the rotor or simply remain on the pump side. (In the devices tested in part II, the holes do not exist; thus, the pump flow exits on the pump side.) In order to separate the two flows and prevent flooding in undesired areas, seals are installed on the turbine and pump side of the rotor. Bearings support the rotor axially and laterally by providing pressurized flows to create restoring forces on the rotor. Pressurized gas flows come from the bottom side and enter into the narrow gap surrounding the rotor, which works as a JB. Asymmetric flow boundary development around the JB contributes to lateral balancing of the rotor when it is perturbed of center (“JB flow”). Additional externally pressurized flows are provided through orifices in the TBs to create a force under the rotor acting against the pressure force in the turbine (“auxiliary TB flow”).
and “main TB flow” in Fig. 2). The TB system consists of a main TB and an auxiliary TB. The main TB aims to provide most of the thrust from the pump side for axial balance. The auxiliary TB is intended to give extra force to complement the main TB and facilitate testing. It also mimics the electric generator load by consuming a large portion of the turbine work through viscous drag over its large surface area. The speed bump in Fig. 2 is for measuring the rotational speed of the rotor. A more detailed description of the components will be the subject of the following sections.

To design this device, low-order analytical models of the components were developed: 1) a turbine model based on mean line analysis with loss correlations extracted from 2-D computational fluid dynamic (CFD) calculations; 2) flow resistance models for the TBs; and 3) models based on lubrication theory for the pump and seal. To evaluate the design space and achievable performance for each component, preliminary design studies will be presented along with the models. Then, they will be combined to design a microturbopump demo system with specific dimensions to achieve watt-scale power levels. The following sections therefore describe the component models and the system-level design of a demo microturbopump.

IV. MULTISTAGE MICROTURBINE

A turbine converts fluidic energy to mechanical energy by changing the angular momentum of the flow. Fixed stator blades impart swirl to the flow, which is then removed in the rotor blades, imparting a torque on the rotor. This process is repeated through an alternating arrangement of stator and rotor blade rows to increase the torque on the rotor and, therefore, the mechanical power delivered (power is the product of torque times rotation rate). To achieve interesting power density levels, microturbomachinery must operate at similar performance levels (i.e., tangential tip speeds) as traditional large-scale turbomachinery and maintain the highest possible efficiency. Unfortunately, the photolithography-based microfabrication approach does not allow the fabrication of truly 3-D turbomachinery, thereby limiting the design freedom often required to create the most efficient flow fields. In order to create aerodynamic profiles, the blades are best defined in the plane of the wafer using micrometer-resolution lithography and then etched to create extrudedlike blades extending normal to the substrate.

Since the turbine blades are formed in a single common etch step, all blades are constrained to a uniform height (i.e., span). Fig. 3 shows a typical micromachined turbine, illustrating the constant blade height geometry. Due to this limitation of silicon micromachining and the planar geometry, the design space is significantly different from that for traditional turbomachinery. The main differences are as follows.

1) Each stage (blade row) operates at a different tip speed, proportional to the radius: \( U = \text{rotation rate} \times \text{radial distance} \).

2) The flow path area increases linearly with radius: \( \text{area} = 2\pi \times \text{radial distance} \times \text{height} \), hence imposing area variations across each blade row and from stage to stage.

3) Blade shapes are constrained to 2-D extrudedlike shapes, without any geometric variation along the span, such as twist.

For the direction of the gas flow, the radial outflow configuration was found to be preferable to the traditional inward flow found in macro and other micro radial turbines. Given the aforementioned geometric constraints, radial outflow maintains a more uniform velocity and power distribution across the stages without significant drawback from the adverse centrifugal effects. For turbine design and performance predictions, a mean line analysis based on velocity triangles with loss factors was used. This low-order modeling approach is the basis for a preliminary design of traditional turbomachinery [11].

The detailed modeling of the microturbine is beyond the scope of this paper and therefore described in a separate paper [12]. The main aspects are summarized here for completeness along with the specific design used for the microturbopump.

A. Turbine Modeling Approach

The flow is considered to be compressible (ideal gas) and adiabatic. The process consists of conserving total enthalpy across each stator, rothalpy across each rotor, and mass throughout. Eventually, pressures and temperatures across the turbine should be estimated through the calculation process in order to assess the turbine performance.

A pressure loss coefficient correlation is applied to represent the viscous losses across the blade rows, and a blockage coefficient is used to consider the boundary layer growth and changing passage area. The pressure loss coefficient is used as the turbomachinery loss factor, as is defined in [12]

\[
Y = \frac{P_{01} - P_{02}}{P_{01} - P_2}
\]  
(1)

The correlation for the profile loss of the turbine blades was obtained by 2-D CFD calculations [12] and expressed as a function of Reynolds number based on boundary layer theory [13] as follows:

\[
Y = \frac{C_1}{\sqrt{\text{Re}}} + \frac{C_2}{\text{Re}}
\]  
(2)

where \( C_1 \) and \( C_2 \) are functions of geometry. To account for 3-D loss effects, a correction factor was multiplied to (2), as
suggested in large-scale turbine research [11] and microscale investigations [14].

Blockage is also a factor that diminishes the performance of turbomachinery. The blockage was defined based on previously published CFD results [14] and boundary layer growth theory as follows:

\[ K = 1 - a \sqrt{\Delta r} \]  

(3)

where \( \Delta r \) represents the radial position from the flow inlet and the constant “\( a \)” depends on the geometry and flow condition.

Deviation effects were not included in this model due to the limited data available to parameterize it as the function of the Reynolds number or geometry.

B. Preliminary Design—Multistage Turbine

To date, most of the microscale turbomachines [2], [3] have one stage, i.e., one stator and one rotor pair. However, a single-stage configuration is limited to pressure ratios higher than two for macroscale axial machines or seven for centrifugal impellers, and even less at microscale. However, pressure ratios above these values (> 100) are critical to obtain high Rankine-cycle efficiency. Multiple turbine stages are therefore required to completely expand the flow in a Rankine cycle; hence, a multistage configuration must be developed to implement this cycle at microscale. The configuration adopted here is therefore different from previous microturbines given the goal of creating multiple blade rows in a single plane. The focus in this paper is therefore on implementing the multistage configuration but at modest pressure ratios. Further efforts and new designs such as multispool configuration, which is discussed in [15], are expected to raise the pressure ratio near common levels in conventional large-scale turbines.

In order to investigate the effect of the number of stages on turbine performance, a preliminary design study of multistage turbines on a single disk with radial outflow has been performed. The inlet condition was given as 400 °C, 0.6 MPa, and 24 mg/s of flow rate of steam, which was directly taken from our previous cycle analysis [4]. From this study, several important features for optimal design have been found. First, each stage can produce similar levels of power in the outward flow configuration. Second, the optimal rotation rate, at which highest power production occurs, can be matched for all stages. These two features indicate that load can be well distributed among multiple stages. Third, turbines with more than five stages can operate poorly due to mismatch of power production. At the sixth stage, the power production is much higher than that of other stages due to the excessive expansion of the flow at the optimal rotation rate, while it is lower at other speeds due to poor expansion. Configurations with five stages or less per rotor are preferable. This design study also showed that it can produce a 1-W level of power from each stage for the given flow rate, a total of 5 W, which corresponds to 208 kJ/kg of power density, with the rotor diameter of 1.5 mm.

To evaluate the effect of scale on efficiency, the performance at different Reynolds numbers was also studied with a turbine of different design (four stages, 4-mm diameter, and blade chord length of 100 μm) [12]. The calculations show that the average Reynolds number across blade rows will be less than 1000 for flow velocities of conventional turbines (several hundreds of meters per second). This suggests that the flow will be laminar and that energy loss can be somewhat significant at these scales. Low Reynolds number results in low efficiency, calculated to be on the order of 22% at Re = 200 up to 63% at Re = 700. According to our previous Rankine-cycle analysis, 70% of turbine efficiency will enable the whole Rankine micro power generation system to deliver reasonable levels of power (1–10 W) and system cycle efficiency (1%–10%) [4]. Turbine size should therefore be kept on the millimeter scale, with blade chord of 100 μm or more, and rotational speeds should be maximized. The rotor diameter chosen for the microturbopump device is 4 mm, respecting this minimal-scale criterion and leveraging previous experience with IBs at this size [8], [16]. For further details of the modeling and design study of the turbine, the readers are encouraged to see [12].

V. Spiral Groove Viscous Micropump

In order to deliver a few watts of power production, the device will need only a few tens of milligrams per second of liquid flow rate and about 5–10 of pressure ratio according to the thermal cycle analysis of a Rankine vapor cycle [4]. Rough calculations suggest that the dimensions of a radial flow pump should be about 100 μm in diameter and a couple micrometers in height to satisfy those conditions. Hence, the Reynolds number of the flow in the pump will be much less than one, and viscous forces tend to dominate over inertial effects. By principle, turbomachines operate on inertial effects through angular momentum changes and centrifugal forces, and viscous effects induce losses. A more natural approach at low Re is to leverage the viscous forces to do work on the flow. Therefore, the shear driven approach in a viscous pump is favored over a turbomachinery-based pump. Here, the micropump design is directly inspired from hydrodynamic spiral groove TBs, with a center port added to collect or draw the fluid. The geometry consists of a grooved surface with shallow trenches of depth \( h \), spiraling inward at a constant spiral angle \( \alpha \), parallel to a smooth surface as shown in Fig. 4. As the disk rotates over the grooves, the shear force drags the fluid in the grooves, effectively pressurizing it. The analytical model for design and analysis was mainly taken from the study of Muijderman [17].

A. Viscous Micropump Modeling Approach

Traditionally, spiral groove TBs have been designed and analyzed using the Reynolds equation, which assumes fully developed thin-film viscous flow [17]. This approach is applicable for the microscale viscous pump because of its thin planar shape and low Reynolds number flow. This model is based on some basic assumptions: 1) fully developed viscous flow; 2) incompressible; 3) uniform pressure and density across the thickness of the lubricating film; and 4) constant fluid properties (viscosity).
The pressure rise, torque, and power consumption as a function of geometry, rotational speed, and flow rate are determined based on published work [17] and detailed elsewhere [15].

**Pressure Rise Equation and Thrust Force**

\[
\Delta P = \frac{3\pi\mu\omega}{h_2} (r_{2\text{eff}}^2 - r_{1\text{eff}}^2) g_1
\]

\[-6\mu(1 + \gamma)\pi h_1 h_2 A \ln\left(\frac{r_{2\text{eff}}}{r_{1\text{eff}}}\right)\]

\[F = \frac{3\pi\mu\omega r_2^4}{2h_2^2} \left(1 - \lambda^2\right)^2 g_1 C_s^2\]

\[-\int_{r_{1\text{eff}}}^{r_{2\text{eff}}} \left[\int_{r}^{r_{2\text{eff}}} \frac{12\mu(1 + \gamma)}{\rho h_1 h_2} g_3 \frac{dr}{r} \right] r dr\]

**Torque Equation**

\[\text{Torque}_{\text{pump}} = \frac{\pi\mu\omega r_2^3}{2h_2^2} (1 - \lambda^2) g_2\]

\[-B \left(r_2^2 - r_1^2\right) \frac{\mu\gamma}{\rho h_1 h_2}\]

where \(g_1\), \(g_2\), and \(g_3\) are functions of \(\alpha\), \(\gamma\), and \(H\), which are the angle of spiral groove and ratios of \(a_2\) to \(a_1\) and \(h_2\) to \(h_1\), respectively (see Fig. 4 and the Nomenclature section).

The first term in (4) represents pressure buildup due to viscous drag, while the second term accounts for pressure drop due to the flow through the pump. The thrust force is obtained by integrating the pressure equation from the inner to the outer radius. The torque is calculated by integrating the Couette flow equation over the grooves and ridges.

**B. Micropump Design Optimization**

In order to determine the geometry that provides a desired pressure rise with the least power consumption, an optimization process is needed. According to other studies [17], [18], the configuration is optimized for a spiral angle of \(\alpha = 15^\circ\) and a groove/ridge width ratio of \(\gamma = 1\), independently of the other parameters. Therefore, the optimization process is reduced to a set of four variables as follows:

\[\Delta P = f_1(H, h_2, r_2, \lambda)\]

\[\text{Power} = f_2(H, h_2, r_2, \lambda)\]

The optimization process consists of choosing a rotation rate and then calculating the pressure rise and power for the entire range of design parameters. The optimum design was then found by searching for the design with the least power consumption while delivering a pressure rise in the desired range.

Pumps with low pressure rise (0.6 MPa) and high pressure rise (8 MPa) delivering 24 mg/s of water at \(4 \times 10^5\) rad/s (3.8 million r/min) were designed according to the aforementioned process. Table I and Fig. 5 show the configurations and characteristic curves of the pumps. Their efficiencies were 4.3%, consuming 0.35 and 4.6 W, respectively, with diameters of \(\sim 300\) \(\mu\)m. The optimization process revealed that pumps with the same level of efficiency could be designed for various operating rotation rates. Although the efficiency is quite low, a Rankine-cycle analysis suggests that the power consumption by the pump at the above efficiency will account for only 3% of the turbine power generated, suggesting that the pump efficiency is not critical in this range [4]. This design exercise demonstrates that spiral groove viscous pumps are a promising approach for microscale systems. The efficiency is acceptable over a wide range of design pressures and velocities, the geometry is readily produced with microfabrication, and the scale is compatible with the other components of a mini Rankine-cycle steam turbine.

**C. Centrifugal Effect**

Although the pumping mechanism depends mainly on the viscous effect, centrifugal forces can influence the pump performance [17]. The ratio of the centrifugal force to the viscous force can be expressed as

\[\frac{\text{centrifugal force}}{\text{viscous force}} \propto \frac{r_2^2 \rho \pi r_2^2 h_2}{\mu \frac{\pi r_2^3}{h_2}} = \frac{\omega h_r}{\nu} \]

For the low-pressure-pump configuration proposed earlier, the value of this nondimensional parameter is approximately

\[\frac{\text{centrifugal force}}{\text{viscous force}} \propto \frac{\omega h_r}{\nu} = \frac{4 \times 10^3 \times (0.5 \times 10^{-6})^2}{0.4 \times 10^{-6}} = 25\%\]

which suggests that the centrifugal forces are not necessarily negligible, and should be included in the model.Muijderman
TABLE I

<table>
<thead>
<tr>
<th>Pressure rise (ΔP)</th>
<th>0.6 MPa</th>
<th>8 MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \dot{m} )</td>
<td>24 mg/s</td>
<td>24 mg/s</td>
</tr>
<tr>
<td>Rotation rate (( \dot{\theta} ))</td>
<td>( 4 \times 10^5 ) rad/s</td>
<td>( 4 \times 10^5 ) rad/s</td>
</tr>
<tr>
<td>Angle (( \alpha ))</td>
<td>15°</td>
<td>15°</td>
</tr>
<tr>
<td>Gap between groove and the rotor (( h_b ))</td>
<td>4.61 ( \mu )m</td>
<td>3.12 ( \mu )m</td>
</tr>
<tr>
<td>Gap between ridge and the rotor (( h_d ))</td>
<td>0.6 ( \mu )m</td>
<td>0.5 ( \mu )m</td>
</tr>
<tr>
<td>Inner radius (( r_1 ))</td>
<td>214 ( \mu )m</td>
<td>243 ( \mu )m</td>
</tr>
<tr>
<td>Outer radius (( r_2 ))</td>
<td>275 ( \mu )m</td>
<td>450 ( \mu )m</td>
</tr>
</tbody>
</table>

| Power | 0.35 W | 4.60 W |
| Efficiency | 4.3% | 4.3% |

Fig. 5. Pump characteristic curves generated by the model calculations: (Dashed lines) Pressure rise and (solid lines) efficiency as a function of mass flow rate and speed for the low-pressure design. The unit of the rotation rate is rad/s, and efficiency is expressed as a percentage. (a) Low-pressure viscous pump. (b) High-pressure viscous pump.

[17] suggests that the average tangential velocity, which is half of the rotational velocity, can be used for the approximate pressure rise calculation. Considering this assumption, the model of the viscous pump can be modified to

\[
\Delta P_{\text{mod}} = \Delta P + \frac{1}{8} \rho \Delta (r^2 \omega^2)
\]  

(7)

where the centrifugal term will be positive for outward pumping and negative for inward pumping. Rough estimations suggest that the centrifugal effect is ignorable at low speeds but becomes nonnegligible at high speeds such as a million of revolutions per minutes for pumps with size of several hundreds of micrometers in diameter.

VI. PARTIALLY GROOVED SEAL

Sealing is required to separate the high-pressure fluid from the low-pressure fluid and separate the liquid and gas streams. As shown in Fig. 2, seals are needed at two locations. Without a pumping mechanism on the seal, which could consist of a simple planar annular clearance, the liquid from the pump would flow to the gas side. Although small clearances would restrict the leakage flow rate, the accumulation of liquid in the outer radius under the disk would dramatically increase the viscous drag and reduce the Rankine device efficiency. On the turbine side, flow in the turbine due to leakage from pumped liquid can influence negatively the turbine performance by reducing the efficiency and damaging the blades. Thus, an inward pumping partially grooved seal was designed to prevent liquid leakage, as in Fig. 6.

A. Seal Modeling Approach

The modeling is similar to that of the pump, except for the added flat surface area. The pressure \( P_b \) between the grooved and ungrooved parts is determined by substituting (8) in (4) and replacing \( r_{1,\text{eff}} \) by \( r_b \) in (4); the entire pressure distribution between \( r_2 \) and \( r_1 \) is then determined. Flow in the ungrooved
area is considered as the Couette flow in the tangential direction and fully developed Poiseuille flow radially

\[
\dot{m} = \frac{\pi \rho h_2^3 (P_b - P_1)}{6 \mu \ln(r_b/r_1)}
\]

(8)

\[
P_1 - P_2 = \Delta P_{\text{grooved}} + \Delta P_{\text{ungrooved}}
\]

(9)

\[
\Delta P_{\text{ungrooved}} = -\frac{6 \dot{m} \mu \ln(r_b/r_1)}{\pi \rho h_2^3}.
\]

(10)

Instead of deriving the closed solution of the flow rate and pressure from the equations, an iterative computational technique was used by changing the flow rate of (8) and substituting it into (9) until \(P_1\) matches the pump exit pressure. Once the pressures are defined, the thrust force formed on the seal is calculated by integrating the pressure distribution over the grooved and ungrooved areas as in (5)

\[
F_{\text{seal}} = F_{\text{grooved}} + F_{\text{ungrooved}}
\]

(11)

where the force on the grooved area is defined by simply substituting \(r_1\) and \(\lambda\) with \(r_b\) and \(\lambda_b\) in (5). Force in the ungrooved area is

\[
F_{\text{ungrooved}} = -\frac{6 \dot{m} \mu \ln(r_b/r_1)}{\pi \rho h_2^3} (r_b - r_1) - r_1 \ln(r_b/r_1) + P_b (r_b - r_1).
\]

(12)

Considering the ungrooved area, the frictional torque of the seal is

\[
T_{\text{seal}} = T_{\text{grooved}} + \frac{\pi \mu \omega (r_b^4 - r_1^4)}{2h_2}
\]

(13)

where \(T_{\text{grooved}}\) is calculated from (6) by substituting \(r_1\) and \(\lambda\) with \(r_b\) and \(\lambda_b\).

As mentioned before, this model is based on the assumption that the working fluid is incompressible, which is only valid for low-pressure operation with a gas. However, based on Muijderman’s argument [17], compressibility gives a benefit in pressurization, so using the current model is expected to lead to a more robust and conservative seal once implemented.

According to a preliminary design study, a simple planar annular clearance, whose \(r_1\), \(r_2\), and \(h_2\) are 650, 900, and 0.5 \(\mu\)m, respectively, will have 0.1 \(\mu\)g/s of water leakage from the pump to the gas side for 5 atm of differential pressure across the clearance. In contrast, there will be 2.0 \(\mu\)g/s of gas leakage to the pump flow for a partially grooved seal at the same condition. The leaked gas corresponds to 0.05% of the pump flow in mass and is expected to have negligible influence to the pump performance.

**B. Preliminary Design**

A seal, along with a spiral groove viscous pump with outward pumping, was designed to assess the feasibility of the concept. The seal surrounds the pump with a 100-\(\mu\)m clearance between them (see Table II), which is required as the pump outlet channel. Along the pump pressure rise curve, the pump flow rate increases linearly with the rotational speed up to 122 mg/s at 1.5 million r/min, and centrifugal effects were taken into account for the pressure rise in the pump. Fig. 7 shows that the seal pressure rise exceeds that of the pump at any operating speed and the seal can effectively block the water from flooding the back side of the rotor.

**C. Extra Forces at the Interface of Two Fluids (Capillary and Centrifugal Effect)**

According to the design analysis in Section VI-B, the gas pressure will be slightly higher than the liquid pressure for the operating speed range (0–1.5 million r/min). However, since the lighter fluid (gas) holds back the heavier fluid (liquid), the seal can be subject to the Rayleigh–Taylor instability, which happens at the interface of two fluids under different body forces. In order to estimate this effect, we consider two extra forces incurred at the boundary of the pump and the seal: capillary force from the seal gap and centrifugal force developed in the
liquid between the pump exit and the seal. The ratio of the centrifugal force to the capillary force can be expressed as follows:

$$\frac{\text{centrifugal force}}{\text{capillary force}} \propto \frac{\Delta P \left(r^2 \omega^2\right) 2\pi r h}{\frac{2\Delta P}{\pi^2} 2\pi r h} = \frac{\rho \Delta \left(r^2 \omega^2\right) h}{16\Gamma}$$

where we used the centrifugal term in (7) to represent the centrifugal force. This nondimensional term is used to determine the force exerted from the small gap region with the rotation rate while the capillary force is constant. Based on the design dimensions in Section VI-B, the two forces are balanced at the rotation rate of 1.35 million r/min, above which the Rayleigh–Taylor instability can occur. This simple estimation suggests that the force by surface tension at the small gap of the seal is not trivial and contributes to suppressing the Rayleigh–Taylor instability up to fairly high speeds.

VII. TB

The axial balance system consists of a main TB and an auxiliary TB to support the rotor against the thrust force on the turbine side (Fig. 2). In the main TB, the flow is extracted from the turbine inlet through orifices in the rotor. The auxiliary TB flow, however, is supplied from an external source. This additional degree of freedom will allow independent control of axial position by regulating the auxiliary TB pressure during testing. Because the turbopump under development does not have the generator to consume the power produced by the turbine, the auxiliary TB surface is also used as a viscous load that stems from the shear in the small bearing gap. The modeling approach of the TBs is similar to that of Lin [8], but the configuration is customized for the current application. Unlike similar types of TB systems previously developed [3], [8], the current design approach aims at developing TBs only on one side of the rotor in order to reduce device complexity.

The hydrostatic TB flow is usually analyzed using an electrical circuit analogy, where the pressure drop can be expressed as the product of mass flow and flow resistance. Fig. 8 shows a simplified schematic of the TB flow and working principle. When the flow goes through orifices in the rotor, pressure drops from $P_{in}$ to $P_{TB}$ by the orifice resistance $R_o$. After coming out of the orifice, the flow spreads out radially through the large gap at the pump side of the rotor, where the pressure distribution is considered uniform. Pressure then drops from $P_{TB}$ to $P_{out}$ as it flows through the small gap region with flow resistance $R_b$. When the rotor moves axially, the change in gap affects $R_b$, which regulates the flow rate and, consequently, the intermediate pressure $P_{TB}$ (similar to a voltage divider). This pressure change provides a restoring thrust force on the rotor to maintain the axial balance.

Based on the aforementioned concept and the assumption of laminar incompressible flow, the thrust force of the bearing can be expressed as follows:

$$P_{TB} = P_{in} - \dot{m} \frac{128\mu L}{\pi d^3 \rho N}$$  \hspace{1cm} (14)

$$F_1 = \pi \left(r_2^2 - r_1^2\right) P_{TB}$$  \hspace{1cm} (15)

$$F_2 = \frac{12\mu}{\rho \times g} \dot{m} \left( -\frac{1}{2} \left( r_3^2 \ln r_2 - r_4^2 \ln r_2 \right) \right.\left. + \left( \frac{1}{2} + \ln r_2 \right) \frac{r_2^2 - r_1^2}{2} \right)$$  \hspace{1cm} (16)

$$F = F_1 + F_2$$  \hspace{1cm} (17)

where $F_1$ is the force exerted from the large gap area and $F_2$ is the force from the small gap area.

To properly represent the behavior of a real bearing, multiple flow resistances must be accounted for, such as the entrance region of the orifice $(R_1)$, the orifices $(R_2)$, the exit and turning $(R_3)$, the large clearance channel $(R_4)$, the contraction region $(R_5)$, the restrictor $(R_b)$, and the expansion region $(R_7)$, as shown in Fig. 9.

The flow modeling is based on simple fully developed compressible flow in the orifices, channel, and restrictor, combined with experimental correlations for inertial pressure drop in the contractions, expansions, and bends. The modeling approach is similar to Deux (refer to [20] for the detail of force and
flow rate equations), but the main difference is the addition of the restrictor, which requires pressure drop models of the contraction and expansion region \((R_5\text{ and } R_7\text{ in Fig. 9})\). Unfortunately, most contraction and expansion correlations are not for the specific geometry found here and are typically for larger Reynolds numbers \((\text{Re} > 100)\). Given the situation, numerical calculations were used to get the pressure loss data in the restrictor. CFD calculations were performed with the assumption of laminar, incompressible, and 2-D flow using a commercial code (refer to [15] for complete details).

The pressure loss coefficients \(K\)'s for contraction and expansion are calculated based on the following equation:

\[
\Delta P_0 = K(P_0 - P)
\]

where the subscript 0 represents a total (stagnation) property. Using the results of CFD, new correlations of pressure loss coefficient of contraction and expansion were defined. The equations normally should be a function of Reynolds number and the aspect ratio of gap to channel height. The computation was performed for various aspect ratios, and it was found to have very little effect, as long as it is less than one-sixth. Therefore, the correlations can be assumed to be functions of \(\text{Re}\) only, and they are expressed as follows (see Table III for the coefficients):

\[
K_{\text{cont}} = \frac{a_1}{\text{Re}} + \frac{a_2}{\ln(\text{Re})} + \frac{a_3}{\ln(\text{Re})^2} + \frac{a_4}{\ln(\text{Re})^3} + a_5 \times \text{Re} \quad (2 < \text{Re} < 100)
\]

\[
K_{\text{exp}} = \frac{A}{\text{Re}} \quad (\text{Re} \leq 2).
\]

### B. Auxiliary TB

As described in Section VII, the auxiliary TB will be located at the edge of the rotor, on the pump side. The benefit of this location is that it gives a higher restoring torque when tilting happens compared to an inner position. This bearing will be used to float the rotor independently of the turbine pressure, such as during start-up, and to provide stiffness.

### C. Preliminary Design of TBs

A TB system was designed along with a turbine in order to assess its performance. The nominal gaps of both the main and auxiliary TBs are designed to be around 1 \(\mu\text{m}\), considering the dissipation of turbine power at the auxiliary TB surface. The turbine thrust force is calculated for a turbine with 4 mm in diameter and four-stage configuration at the same inlet flow condition. The working fluid is steam for all the components. The specific geometry and the inlet condition for this baseline design are shown in Table IV. Fig. 11 shows that the forces between the turbine side and the bearing side are balanced at a bearing gap of 1.3 \(\mu\text{m}\). According to the modeling, for a force perturbation of 0.2 N (approximately 10\% of the turbine pressure force), the rotor moves only 0.2 \(\mu\text{m}\) within the balanced position. The stiffness, which is calculated by differentiating the net force with respect to the bearing gap, is about 1 N/\(\mu\text{m}\) and is maximized when the rotor is balanced axially. This analysis shows that the TB system is optimally designed and can provide enough force for practical operation of the rotor according to a previous work with a similar configuration [8], [16].

### VIII. Hydrostatic JB

For high-speed operation of silicon microturbomachinery, hydrostatic gas lubricated JBs have been studied and
TABLE IV
GEOMETRY OF THE TBs AND INLET CONDITION

<table>
<thead>
<tr>
<th></th>
<th>Main</th>
<th>Auxiliary</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>TB</td>
<td>TB</td>
</tr>
<tr>
<td>Inlet Temperature</td>
<td>200 °C</td>
<td>200 °C</td>
</tr>
<tr>
<td>Inlet Pressure</td>
<td>4 atm</td>
<td>4 atm</td>
</tr>
<tr>
<td>Diameter of the orifice (d)</td>
<td>20 μm</td>
<td>10 μm</td>
</tr>
<tr>
<td>Radius from center to orifice (r1)</td>
<td>1 mm</td>
<td>1.88 mm</td>
</tr>
<tr>
<td>Inner radius of the restrictor (r2)</td>
<td>1.7 mm</td>
<td>1.81 mm</td>
</tr>
<tr>
<td>Outer radius of the restrictor (r3)</td>
<td>1.71 mm</td>
<td>1.94 mm</td>
</tr>
<tr>
<td>TB gap (g)</td>
<td>1.3 μm</td>
<td>1.3 μm</td>
</tr>
</tbody>
</table>

Fig. 11. Calculated thrust force balance between the turbine and TBs.

successfully demonstrated [16]. The operating principle consists of driving flow axially along the JB gap, from the backside to the turbine side of the rotor. The axial pressure profile along the sidewall of the rotor is not necessarily linear due to inertial and entry losses; hence, it depends on the local gap and Reynolds number. As the rotor moves off center, the JB gap becomes circumferentially nonuniform, inducing nonuniform pressure forces. Since smaller clearances result in lower inertial and entry losses, local pressure forces are higher than in the large clearance sections, yielding a net recentering force. Fréchette et al. [16] demonstrated a gas lubricated hydrostatic JB on the periphery of 4-mm-diameter rotors with 0.3-mm axial length. The configuration showed successful operation at high rotational speeds (up to 303-m/s tip speed or 1.4 million r/min). For the current application, a similar configuration was used, leveraging an experimentally proven configuration.

IX. DESIGN OF A DEMO MICROTURBPUMP

A demo microturbopump system was designed using the models presented earlier. The design parameters are basically specified based on two types of balances: power and axial force balances. The rotational speed of the rotor will be such that the power produced by the turbine is equal to the sum of the power dissipated in the other components. In addition, the rotor will position itself axially such that the pressure forces on the turbine side of the rotor balance those on the pump side. For the system design, models corresponding to the components are combined together to estimate the powers and forces in order to find the balanced steady-state operating conditions (speed and axial position). This process is repeated by changing the design parameters until the optimum or desired performance is obtained.

A. Power Balance

The mechanical power to drive the rotor is produced by the turbine and is dissipated by the pump, bearings, and seals [see (20)]. The reduction of total enthalpy of the flow across the turbine will be converted into turbine power production as in (21). Dissipated powers in the JB and TBs are modeled from the assumption of the Couette flows in the bearing clearances [see (22)–(24)]. Powers consumed or dissipated by the pump and seal are defined from the products of the torque and rotation rate, as shown in (25) and (26). In the modeling, eccentricity and tilting were not considered on the assumption that they are all at stable operating conditions with the rotor centered.

\[ \dot{W}_{\text{turbine}} = \dot{W}_{\text{viscous drag}} = \dot{W}_{\text{JB}} + \dot{W}_{\text{main TB}} + \dot{W}_{\text{aux TB}} + \dot{W}_{\text{pump}} + \dot{W}_{\text{seal}} \]
\[ \dot{W}_{\text{turbine}} = \dot{m}(h_{0,\text{inlet}} - h_{0,\text{outlet}}) \]
\[ \dot{W}_{\text{JB}} = \frac{2\pi \mu_{\text{gas}} \omega^2 l_{\text{JB}} r_{3,\text{B}}^3}{c} \]
\[ \dot{W}_{\text{main TB}} = \frac{\pi \mu_{\text{gas}} \omega^2 (r_{4,\text{main TB},2}^4 - r_{4,\text{main TB},1}^4)}{2g_{\text{main TB}}} \]
\[ \dot{W}_{\text{aux TB}} = \frac{\pi \mu_{\text{gas}} \omega^2 (r_{4,\text{aux TB},2}^4 - r_{4,\text{aux TB},1}^4)}{2g_{\text{aux TB}}} \]
\[ \dot{W}_{\text{pump}} = T_{\text{pump}} \times \omega \]
\[ \dot{W}_{\text{seal}} = T_{\text{seal}} \times \omega. \]

The power dissipated from viscous drag (pump, bearing, and seal) between the rotor and the stationary surfaces is proportional to the rotation rate squared \((\omega^2)\), as shown by the dashed lines in Fig. 12. The turbine power production curve rises up at low speed to reach a maximum and goes down at higher speeds. The rotation rate of the rotor is defined at the point where the two curves intersect. Among the design parameters, the gap and surface area of the components directly affect the viscous drag. When the gaps are smaller and the areas are larger, more power is dissipated, and the rotor spins at lower speed for a given turbine geometry and differential pressure. The opposite happens for larger gaps and smaller areas (Fig. 12). However, higher speed does not always guarantee more power production, as shown in Fig. 12. The optimum design point is where the two curves meet to produce the maximum power from the turbine.
B. Axial Force Balance

The turbine, pump exit area, and top seal impose a downward thrust force from the turbine side. The forces from the main TB, auxiliary TB, bottom seal, and pump push the rotor upward from the pump side. Here, the turbine force is calculated by integrating the static pressure distributions on the assumption of linearity between inlet and outlet of each blade row, and the other forces are defined from the model equations given in the previous sections. The scale of the force from each component varies according to the supplied pressure, operational speed, and geometry. The turbine and the main TB are supposed to take the major roles in the axial force balance, mainly due to their large areas. The forces from the seals and pump are incurred hydrodynamically when the rotor rotates at high speed by raising the pressure in the groove and ridge features. The thrust forces from the turbine and TBs are defined hydrostatically by the supplied pressure regardless of the speed. However, these are coupled since higher turbine inlet pressure induces higher rotational speed and proportionally increases the hydrostatic and hydrodynamic thrust forces simultaneously. The axial position of the rotor will be designed to have the maximum stiffness such that the rotor can operate in a stable fashion.

C. Design Specification and Performance

The design point for this first generation demo microturbopump was chosen to be somewhat conservative compared to the requirements for a Rankine cycle to limit development risks. Table VI shows the resulting design of the test device for an inlet condition of 200 °C and 3-atm steam to the turbine and auxiliary TB, and an outlet at atmospheric pressure. The predicted operating condition of each component is presented in Table V. Steam with the aforementioned inlet condition for this test device is chosen because it has close viscosity value with air at atmospheric temperature and the same pressure. The turbine analysis showed that the flow behaviors of the two fluids were very similar in a wide range of operation. Therefore, the design can be tested with air to verify the turbine model before testing with steam. Many technical considerations were made in the design process, including the following: 1) The turbine blade angles were designed considering the induced flow deflection at the leading edge and deviation at the trailing edge [21] (a few degrees were added to the originally designed angles to address them); 2) the groove number of the pump and seals were defined to make sure that the circumferential length of one groove is more than 20 times of the groove depth so that the model from lubrication theory can be valid; 3) the pump design was optimized considering all variables, including $\alpha$ and $\gamma$ for best performance; 4) the TBs were designed to have high spring constants to run the rotor at a stable condition; and 5) the rotor diameter was constrained to 4 mm considering turbine efficiency and the previous experience with this rotor scale [16].

Fig. 13 shows the axial force balance on the rotor for the given geometry and conditions, as a function of the axial position noted by the gap between the rotor and the TB surfaces. In these calculations, the rotor axial position is set, and the rotational speed is iterated upon until the power balance is satisfied. This calculation is then repeated for various bearing gaps, i.e., rotor axial positions. The actual rotor position is then defined...
TABLE VI

<table>
<thead>
<tr>
<th>Dimensions of Demo Microturbopump Device</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Turbine</strong></td>
</tr>
<tr>
<td>Outer radius</td>
</tr>
<tr>
<td>Number of stages</td>
</tr>
<tr>
<td>Solidity</td>
</tr>
<tr>
<td>Chord</td>
</tr>
<tr>
<td>Clearance between blade rows</td>
</tr>
<tr>
<td>Blade height</td>
</tr>
<tr>
<td>Blade profile</td>
</tr>
<tr>
<td>Blade inlet angle</td>
</tr>
<tr>
<td>Blades exit angle</td>
</tr>
<tr>
<td>Leading edge radius of blades</td>
</tr>
</tbody>
</table>

| **Pump**                                 |
| Radial extent                           | \(r_1: 400 \mu m, r_2: 550 \mu m\) |
| Groove Number                           | 16    |
| Spiral angle (\(\alpha\))              | 16°   |
| Groove depth (\(h_{y-h}\))             | 6 μm  |
| Ratio of ridge to groove width (\(\gamma\)) | 0.3 |
| Design operating gap (\(h_2\))         | 0.5 μm |
| Design operating gap range              | 0-1 μm |

| **Seal**                                 |
| Radial extent of grooved part           | \(r_{y}=700 \mu m\), |
| Groove number                           | 20    |
| Groove depth (\(h_{y-h}\))             | 1.5 μm |
| Circumferential Ratio of ridge to       | 1    |
| Spiral angle (\(\alpha\))              | 16°   |
| Radial extent of ungrooved              | \(r_2=650 \mu m, r_2=700 \mu m\) |
| Design operating gap (\(h_2\))         | 0.5 μm |
| Design operating gap range              | 0-1 μm |

| **Journal Bearing**                     |
| Gap                                     | 20 μm |
| Feed plenum layout                      | 2×120° |
| Length                                  | 400 μm |

| **Main Thrust Bearing**                 |
| Number of capillaries (\(N\))          | 32    |
| Length of capillaries (\(L\))          | 400 μm |
| Diameter of capillaries (\(d\))        | 20 μm |
| Radial position of capillaries (\(r_1\)) | 950 μm |
| Radial extent of restrictor             | \(r_2=1540 \mu m, r_2=1550 \mu m\) |
| Design operating gap (\(g\))           | 1 μm  |
| Design operating gap range              | 0.5-1.5 μm |

| **Auxiliary thrust bearing**            |
| Number of capillaries (\(N\))          | 40    |
| Length of capillaries (\(L\))          | 150 μm |
| Diameter of capillaries (\(d\))        | 10 μm |
| Radial position of capillaries (\(r_1\)) | 1860 μm |
| Radial extent of restrictor             | \(r_2=1750 \mu m, r_2=1975 \mu m\) |
| Design operating gap (\(g\))           | 1 μm  |
| Design operating gap range              | 0.5-1.5 μm |

as the point when the sum of axial forces is zero. According to the calculations, the axial forces on the rotor are balanced at a gap of 1.1 μm, with the spring constant of 2 N/μm. When both the power and axial force are balanced, the rotor is predicted to run at 260 m/s of tip speed or 1.24 million r/min of rotation rate.

Fig. 14 shows the power consumption ratio of each component when the rotor runs at this balanced condition. The power consumed by the pump, which is 20% of the mechanical power, shows that it does not affect the efficiency of the overall device so much considering the low efficiency of the pump. Since the pump was designed to deliver more mass flow and pressure rise than required for the turbine, the power consumption of the pump is higher than that (3%) from cycle analysis mentioned in Section V-B. Fifty percent of the turbine power was dissipated by the auxiliary TB, which would be converted into electrical power by a generator in an actual Rankine device. The mechanical drive efficiency, which is defined as the ratio of useful work (here, sum of pump work and auxiliary TB drag) to actual turbine work, amounts to 71%, which indicates relatively low level of energy waste to viscous drag considering the large surface area to volume ratio at microscale. The conversion efficiency from fluid energy to electric energy would be 17.6%, assuming 50% of generator efficiency.

X. CONCLUSION

Component and system-level modeling approaches have been presented for the design of a microturbopump as part
of a Rankine microturbine power plant-on-a-chip. Analytical models, supported by empirical correlations and computational analysis, were developed for the components and used to explore the performance levels that can be expected. The models were integrated to enable the demo microturbopump system design while satisfying power and force balance on the free-floating rotor. From the system design analysis, a demo microturbopump was designed to produce 4.7 W of mechanical power from the turbine with the rotor spinning at 260 m/s of tip speed while pressurizing water by 4.5 atm at a flow rate of 100 mg/s. The mechanical drive efficiency of the turbopump is predicted to be 71%, which shows that a microscale mechanical power generation system can be designed without significant energy loss by viscous drag. Furthermore, a spiral groove viscous micropump appears to meet the high-flow-rate and high-pressure requirements with acceptable efficiency. Using a similar approach, seals can also be designed to contain the high-pressure liquid in the pump zone. Overall, this design study suggests that viable device configurations for a microturbopump are possible, as will be experimentally demonstrated in the second part of this two-part paper.

It is anticipated that the turbopump components and microsystem developed herein cannot only be beneficial for micro power generation but also for other mechanical, chemical, and biological applications such as high-pressure and low-volume liquid delivery and cooling with liquid or two-phase flows.

REFERENCES


Changgu Lee received the B.S. and M.S. degrees in mechanical engineering from Hanyang University, Seoul, Korea, in 1995 and 1997, respectively, and the Ph.D. degree in mechanical engineering from Columbia University, New York, NY, in 2006. He was a Postdoctoral Researcher with Columbia University between 2006 and 2010 in the Department of Mechanical Engineering under the supervision of Prof. James Hone. Since September 2010, he has been an Assistant Professor with the Department of Mechanical Engineering, Sungkyunkwan University, Suwon, Korea. His current research interest is in characterization of mechanical and physical properties of 2-D nanomaterials such as graphene and hexagonal boron nitride and their applications. He has measured mechanical properties of graphene and characterized tribological properties of 2-D materials and investigated optical and electronic properties of hexagonal boron nitride and molybdenum disulfide.

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Dr. Fréchette is a member of the American Society of Mechanical Engineers.