Experimental and Computational Investigation of a MEMS-Based Boiler for Waste Heat Recovery

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Abstract

Thermodynamically limited processes make waste heat abundant in availability. An Organic Rankine Cycle (ORC) steam powered micro system designed to scavenge waste heat from various sources (transportation, industries or solar) is presented. The key boiler component is fabricated and characterized in this work. The system design has been inspired by the various efforts implemented in development of micro heat recovery devices and engines. The complete system consists of three individual micro components 1) boiler, 2) free piston expander and 3) superheater. Specifically, design, fabrication techniques, test setup and results of the miniaturized boiler are presented in this paper. A key design feature of the boiler is the inclusion of capillary channels for fluid flow from the surrounding reservoirs to the heated area. The pressurized steam is created by the boiler as a result of phase transformation of the working fluid. This pressurized steam can be utilized to drive another MEMS device (PZT membranes, turbines, thermoelectric, etc.) to

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Preprint submitted to Energy Conversion and Management

April 22, 2015

generate power. In this upgraded boiler design, radial capillary channels and a thin film glass steamdome were considered to improve the operating efficiency. These inclusions enhanced capillary flow, energy absorption via phase change, mass flow rate and operating pressure. The power inputs of 1.8 W and 2.7 W were selected to simulate and characterize the boiler performance based on real-world waste heat source temperatures. For these power inputs, the maximum power absorption efficiency demonstrated by the boiler via phase change of the working fluid was approximately 88%. The peak operating pressure demonstrated by the boiler was 8.5 kPa. These thermally efficient characteristics of the boiler make it a potential future device for waste heat scavenging.

Keywords: Microboiler, Waste heat, Organic Rankine Cycle, Steam engine, Capillary channels

1. Introduction

In recent years, there has been increasing interest in development of micro-devices and engines to recover waste heat. The reason is its free availability as well as abundance. Such availability is due to the inevitable losses associated with thermodynamically limited processes. In a typical Otto cycle for instance, out of the total energy consumed by the system, only one third of the energy is converted into useful work and the rest is discharged to the surrounding as a waste heat [1]. For a spark ignition 1.4 liter internal combustion engine (ICE) with the thermal efficiency of 15% to 32%, energy

lost through the radiator is 1.7 kW to 45 kW and through exhaust gas is 4.6 kW to 120 kW [1, 2]. Likewise in industries, the estimated losses are in the range of 20% to 50% [3]. These losses could be minimized either by improving the efficiency of the system itself or via installation of secondary waste heat recovery system. The recovery system uses the losses to generate mechanical or electrical work to enhance the efficiency of the primary system [4]. Such notion of heat recovery is utilized in the microdevice presented in this paper.

The possibility of heat recovery using a microdevice was first validated by Muntz et al. in 1991 [5]. The study presented attractive characteristics of a pressure driven microdevice for power generation. Further, the author noted that the pressure driven microdevices required additional study. Since then several different microscale dynamic heat engine types have been investigated for waste heat recovery. Those are micro-sized gas turbines [6, 7] and rotary engines [8, 9], free piston engines [10, 11], and membrane type actuators for power generation [12]. The Brayton cycle based gas turbine was first proposed by Episein et. al.. Anticipated micro-device consisted of eight microscale components to fabricate the fully functional system. The power production by this device was calculated to be in the range of 10-100 W [7]. In recognition to this work, several other researchers initiated investigating of rotatory parts for power generation based on Rankine cycle steam engines [8, 13, 14]. The steam engines were further tailored to operate from low temperature waste heat sources by utilizing low boiling point refrigerants

33 [15, 16, 17]. These micro-devices were called organic Rankine cycle (ORC)
34 steam engines. The ORC based engines were comparatively less complex
35 in design and more economical in terms of cost and maintenance [18]. A
36 Wankel-type ORC engine with rotatory design was studied as well [19].

37 Apart from rotary engines, free piston Sterling engines were also inves-
38 tigated [20]. Metha et. al. presented a comparison of a sterling engine
39 for waste heat recovery using two different ORC working fluids [21]. Simi-
40 larly, performance of the low temperature solar powered differential sterling
41 engine was also investigated. The study indicated a potentially attractive fu-
42 ture low temperature engine [22]. In contrast to rotating and sliding engines,
43 Washington State University introduced a P3 heat engine for power gener-
44 ation [12, 23]. The engine was heated by an external heat source causing
45 phase transformation of the working fluid to gas. The expansion of the fluid
46 deflected the piezoelectric film attached to the silicon membrane produc-
47 ing power. Similar to this, Saturation Phase-change Internal Carnot Engine
48 (SPICE) was studied as well [24]. With consideration of all these efforts in
49 development of microdevices and engines for waste heat recovery, a system
50 capable of scavenging waste heat is presented in this paper.

51 The system currently under investigation consists of three components:
52 1) boiler, 2) free piston expander (FPE), and 3) superheater. Figure 1 shows
53 the ORC system in completion. The boiler is the particular emphasis of the
54 work presented. The boiler consists of microfabricated capillary channels
55 and steamdome. The capillary channel provides continuous supply of the

56 fluid to the heated areas via capillary action, which would otherwise only
 57 be possible using a pump. The boilerplate captures the waste heat initiating
 58 phase change of the working fluid to vapor. The resulting steam traverses the
 59 superheater beneath the boiler. The pressurized superheated steam is then
 60 introduced to the FPE unit. The FPE constitutes the power output from
 61 the system and may produce either mechanical or electrical output depending
 62 on construction and need [25, 26, 27]. Following expansion in the FPE, the
 63 working fluid is cooled and returns to the boiler.

Figure 1: A waste heat scavenging system: the heat source, the steamdome covering, the boilerplate, the FPE and the superheater

64 The work presented in this paper is focused on microscale boiler develop-
 65 ment area. The design and the microfabrication techniques utilized to man-
 66 ufacture the boiler are explained. The detail experimental plans and results
 67 obtained from the fabricated boiler are discussed in the flowing sections.

2. Experimental Materials and Method

The microboiler tested and characterized in this work was based on the design of multiple capillary channels that accessed working fluid reservoirs. The channels transported fluid across the heated boilerplate surface where phase change was initiated. Once boiled, the working fluid was collected in a central steam dome with single outlet. In real-world operation, the outlet can be connected to other small-scale devices like a mechanical expander for final power output and delivery [25, 26, 27]. Figure 2 shows the basic operation and arrangement of the microboiler.

Figure 2: Working demonstration of the waste heat scavenging system

Capillary channels themselves were designed based on prior experimental work [28, 29, 30] as well as expectations from open channel models [29, 31]. Two components were brought together to form the boiler. The boiler design consisted of (1) a boilerplate base with radially separated capillary channels and (2), a steam collecting dome. The overall dimension of the boiler was 35 mm by 35 mm by 1 mm. Photolithographic masks were designed for each microboiler part to transform the design into the working prototype (see Figure

3). A third and final mask was designed to etch inlet and outlet holes into the steamdome top. This avoided the need for any manual drill use. Each part was fabricated independently and then assembled. Additional components were then added to allow for full characterization of the completed boiler.

Figure 3: Boilerplate mask with an enlarged view of the boiler channel design

2.1. Fabrication

A 4 inch type < 100 > silicon wafer was utilized to fabricate the base boilerplate. First, 200 nm thick aluminum was coated on one side of the wafer. This coating served as an etch stop in subsequent fabrication steps. The aluminum was then spin coated with S1813 photoresist (PR) and soft baked at 115°C. The soft baked wafer was exposed and developed using MF-319 developer for about 30 seconds. The developed wafer was then soaked in an aluminum etchant. The aluminum at the photolithographic exposed area was then stripped leaving silicon features beneath. ICP (Ion Couple Plasma) etching was performed on the silicon features to yield channels and the surrounding reservoirs. The depth and width of the capillary channels

were $190\text{ }\mu\text{m}$ and $100\text{ }\mu\text{m}$ respectively. An image of one such fabricated boilerplate is shown in Figure 4.

Figure 4: ICP etched boiler plate with a SEM image showcasing the enlarged channels

After the fabrication of the boilerplate, a glass wafer of similar dimension to the silicon wafer was selected for the steamdome fabrication. The glass was preferred as a material of fabrication because of its insulating properties. The glass also represented significant mass reduction as compared to prior work [28, 30]. This reduced the steamdome thermal impact as part of boiler operation.

The glass wafer was coated with double layers of gold ($2000\text{\AA}$) atop a chrome layer ($200\text{\AA}$) on both sides of the wafer. This coating served as the etch stop for later processing steps, similar to the aluminum coating in boilerplate fabrication. The wafer was spin coated with S1813 PR on the both sides and soft baked for about 90 seconds at 115°C . The soft baked wafer was exposed and developed to form the gold-chrome covered steamdome features on the front side of the glass wafer. The back side of the glass wafer was aligned to the front side using back side alignment (BSA) technique to form

115 the inlet and the outlet holes. The aligned back side of the wafer was exposed
 116 and developed. The wafer was then soaked in gold and chrome etchant to
 117 remove the gold-chrome layer on the exposed surface. Finally, glass features
 118 that defined the specific shape and design of the steamdome were etched
 119 using an aqueous solution of hydrofluoric acid with an etch rate of $5 \frac{\mu m}{min}$. A
 120 diagram of the final steamdome design is shown in Figure 5.

Figure 5: Diagram of fabricated glass steamdome

121 Each fabricated part was anodically bonded together to form the com-
 122 pleted microboiler (see Figure 6). The bonding setup consisted of a hotplate
 123 and a pair of aluminum plates connected to a voltage supply. The individual
 124 parts were cleaned utilizing RCA-I and RCA-II techniques prior to the bond-
 125 ing process [32, 33]. This removed any organic and inorganic contaminations
 126 attached to the bonding surface. The boilerplate and the glass steamdome
 127 were held in place using a pair of aluminum supports. The entire apparatus
 128 was then placed atop the hotplate. The hot plate was preheated to 400°C

129 and a voltage of 400V was supplied with the positive terminal connected to
130 the silicon side [33]. The apparatus was visually inspected at 5 minute in-
131 tervals to assess bonding progress. Relying on this verification, the voltage
132 remained at 400V or was increased up to 700V to enhance bonding qual-
133 ity. After approximately 15-30 minutes, the anodically bonded boiler was
134 obtained. Enclosed volumes for the working fluid and the steam collecting
135 chamber were formed by bonding the boilerplate with the steamdome (see
136 Figure 2). The boiler was now ready for the test setup and characterization.

Figure 6: Top-down view of the anodically microboiler with steamdome and base

137 2.2. Test Setup

138 A number of different components were added to the completed boiler to
139 allow for testing and characterization. These included fluid inlet and outlets,
140 thermocouples (TC), pressure transducers, and heat flux sensors (HFS). For
141 outlet from the boiler, a brass nipple fitting (supplied by McMaster-Carr)
142 was attached to the central exit hole atop the steam collecting chamber. T

143 shaped tubing was fitted to the brass nipple and then properly sealed using
144 epoxy to avoid any leakage of vapor exiting the steamdome during operation.
145 A pressure sensor type P600 (supplied by KAVLICO) was attached to the
146 opening in line with the exit to monitor the microboiler operating pressure.
147 Micro tubing (supplied by IDEX Health and Science LLC) approximately 8
148 mm long was fitted and sealed to the other opening end of the T-tube. This
149 served as an outlet to the microboiler.

150 For the fluid inlet to the microboiler, micro tubing with a pressure sen-
151 sor combined in-line was inserted into the inlet hole atop the reservoirs and
152 epoxied in place to avoid fluid leakage during tests. The sensor monitored
153 any back pressure exerted by the microboiler to the inlet flow. A resistance
154 heater type KHLV-101/P (Omega Engineering Inc.) was used for the heat
155 addition to the microboiler system. The heater was connected to a DC power
156 supply (BK Precision). The supply was capable of reading both voltage and
157 current in a heater connected circuit. This allowed determination of power
158 supplied to the microboiler during testing. K-type thermocouples (Omega
159 Engineering Inc.) were incorporated in the microboiler to monitor the tem-
160 perature of the steam collecting chamber, the central part of the channels, the
161 surrounding reservoirs and the outer surface of the steamdome. Additional
162 thermocouples (TCs) were attached at the bottom of the microboiler to mea-
163 sure input or heater temperature. All attached sensors were connected to a
164 data acquisition system (supplied by National Instruments) and then linked
165 to a computer via LabVIEW SignalExpress software to automate recording.

166 Figure 7 shows the microboiler with additional components included. Input
167 powers of 1.8 W and 2.7 W were supplied to the heater for each series of tests.
168 These powers were selected by comparing the resulting temperature of the
169 heater with the waste heat temperature available from a typical 4 cylinder
170 Otto cycle engine, a potential source for waste heat collection. Temperatures
171 can range from approximately 600°C (exhaust) to 130°C (radiator) [34]. The
172 temperatures reflected by the heater for the given power inputs were in this
173 range and thus well suited for these investigations.

Figure 7: Microboiler test setup placed atop the balsa wood

174 Three independent experiments were performed on the microboiler to
175 fully understand the working capability of the boiler. The experiments were
176 designated as: Dry, Atmospheric and Pressurized. Independent of test type,
177 the boiler was first allowed to heat to steady state prior to data collection.
178 This was verified by monitoring of the operating temperature profiles of the
179 boiler during warm-up. Figure 8 shows a typical temperature recording for
180 a dry test. Steady state was achieved after approximately 12.5 minutes.

181 A detailed explanation of all test setups and techniques are presented in
 182 the following sections. In addition, a detailed finite-difference model was
 183 generated to predict mass flow rate expectations of the boiler in operation.

Figure 8: Typical operating temperatures in dry tests showing steady state

184 2.2.1. Dry Test

185 The Dry test was performed without supplying fluid to the boiler for each
 186 power input. The heat supply component consisted of a heat flux sensor
 187 (HFS) sandwiched between the two heaters. The heat supply component
 188 was then attached to the boiler (see Figure 9). Equal powers were supplied
 189 to the heaters and the HFS output was recorded. The HFS demonstrated
 190 a zero heat flux output. This indicated that there existed no temperature
 191 gradient between the top heater bottom surface and the bottom heater top
 192 surface allowing heat to flow in upward direction towards the boiler.

193 The experiment was conducted until a steady state temperature profile

194 was achieved. The temperature data from the TCs were recorded. Tem-
 195 perature data allowed evaluation of temperatures at various locations of the
 196 microboiler for both levels of power input. Temperature readings were uti-
 197 lized for subsequent model validation work (Section 2.2.1) which was utilized
 198 to predict the available heat flux across the capillary channels for the phase
 199 change of fluid in the Wet tests (Atmospheric and Pressurized). Following
 200 completion of boiler operation without working fluid, the data was utilized
 201 in the modeling effort described in Section 2.2.2.

Figure 9: TCs attached at different locations of the microboiler setup

202 2.2.2. Thermal Modeling

203 A computational model using the Finite Difference Method (FDM) was
 204 generated that replicated the Dry test. The model allowed detailed evalua-
 205 tion of temperatures across the boiler plate and further utilized those temper-
 206 atures to predict the energy available for phase change of the working fluid.
 207 This was utilized to establish expected working fluid consumption capability
 208 for the Wet tests.

Figure 10: Model image of the axis (a), symmetry structures of the uniformly heated radial channels (b), and the enlarged view of the channels

209 A transient heat conduction model was employed assuming uniform tem-
 210 perature distribution across the capillary channels. This was due to the
 211 symmetry structure of the radial channels as shown in Figure 10. A sin-
 212 gle radial channel was considered and the temperature profile was assumed
 213 constant irrespective of change in the theta direction. This reduced the 3-D
 214 problem to a 2-D problem along the $r = x$ and z (thickness) directions. The
 215 model utilized FDM to discretize the heat exchanger into nodal points. The
 216 explicit transient heat conduction equation was solved at each nodal point
 217 to evaluate the temperature using Equation 1.

$$\nabla^2 T = \frac{\rho C_p}{k} \left(\frac{\partial T}{\partial t} \right) \quad (1)$$

218 In Equation 2, T , t , ρ , C_p and k are the temperature, time, density, spe-
 219 cific heat capacity and conductivity, respectively. The material property of
 220 silicon and the dimensions of the wall and the channel were considered. The
 221 convection condition at appropriate nodes was also included in the model.

222 The convection coefficient of approximately $18 \frac{W}{m^2K}$ was considered in the
 223 model. This was determined by introducing known values of supplied heat
 224 flux (q), the resulting temperature via conduction (T) and the ambient tem-
 225 perature ($T_{infinity}$) as recorded in the initial Dry test operation. Equation 2
 226 shows the subsequent calculation of h , the convection coefficient. Moreover,
 227 the calculated value of h was also well in line with the natural convection
 228 range of approximately 5 to $20 \frac{W}{m^2K}$ [13], [14].

$$h = \frac{q}{(T - T_{infinity})} \quad (2)$$

229 Figure 11 shows the 2-D geometry utilized for the model. Appropriate
 230 initial and boundary conditions were applied to replicate the real-world Dry
 231 test. The temperature profile across the model was obtained by converging
 232 iteration based on energy difference ($E_{in} - E_{out}$). It was maintained as low
 233 as micro-Watts for each power input. The nodal temperature generated via
 234 modeling was compared with the Dry test temperature result to validate the
 235 model. After this validation, the amount of thermal energy available from
 236 the channels to deliver to the fluid was evaluated. The heat flux generated
 237 by the model was utilized to calculate the mass feed rate into the boiler. It is
 238 essential to determine the fluid feed rate prior to the Wet tests (Atmospheric
 239 and Pressurized). The mathematical calculation for the maximum mass feed
 240 rate ($\dot{m}$) is given by Equation 3. In Equation 3, the available heat flux
 241 ($P_{available}$) is divided by the working fluid heat of vaporization to yield the

242 mass feed rate. ($P_{available}$) was provided by the model the heat of vaporization
 243 (ΔH_v) was acquired from known properties of the working fluid used in all
 244 Wet tests.

$$\dot{m} = \frac{P_{available}}{\Delta H_v} \quad (3)$$

245 In this work, $3MNOVEC^{TM}HFE - 7200$ was selected as the working
 246 fluid. This was dependent on several things including the temperature at
 247 the boilerplate and other fluid properties like fluid-surface contact angle that
 248 enhanced capillary action, high heat of vaporization, and low environmental
 249 impact. This fluid has a favorable low boiling temperature of 76°C which
 250 was appropriate for the study. It also demonstrated high wettability with
 251 the surface. In addition, it presented high heat of vaporization (ΔH_v) of
 252 about $125 \frac{kJ}{kg}$ and low environmental impact [29]. Due to these favorable
 253 fluid properties, refrigerants has been utilized for boiling heat transfer in
 254 several studies [35, 36]. Considering HFE-7200 as the working fluid the mass
 255 flow rate was determined. This readied boiler for the Wet tests.

Figure 11: A 2-D profile of the 3-D symmetry capillary channel

256 2.2.3. Atmospheric Test

257 In the Atmospheric test, the boiler was operated at atmospheric pressure
258 with working fluid supplied. The working fluid mass exiting the microboiler
259 due to phase change of the supplied working fluid was studied for each indi-
260 vidual power input. The atmospheric test provided insight into the potential
261 of the capillary channels to pump fluid at ambient condition and allowed the
262 determination of a boiler efficiency parameter (energy absorbed via phase
263 change of the fluid (P_{eva})) as defined by Equation 4.

$$Efficiency(\%) = \frac{P_{eva}}{P_{available}} \times 100\% \quad (4)$$

264 Figure 12 shows the test setup with a clamp included that was used to
265 obstruct the exit of the boiler. The clamp was important to subsequent
266 pressurized testing but was not utilized in these atmospheric tests. In these
267 tests, boiler operating pressure was maintained at atmospheric conditions,
268 hence the outlet was left open. A constant heat was supplied from the heater
269 attached to the bottom of the boiler. The temperature and the pressure data
270 were monitored. The boiler was heated for about 12.5 minutes to attain
271 steady state (see Figure 8). After steady state was achieved, the predicted
272 flow rate in the Section 2.1.2 via modeling was supplied.

273 The working fluid was supplied to the boiler via inlet tubing connected
274 to a syringe/infusion pump. The infusion pump was utilized in order to
275 accurately monitor mass flow rates supplied to the boiler. Phase change

Figure 12: Atmospheric and Pressurized test setup

276 occurred on the heated boiler plate as working fluid entered and heated
 277 within the capillary channels. Working fluid exiting the boiler was collected
 278 in a 10cc plastic syringe and cooled using ice cubes surrounding the syringe
 279 for the fixed period of about 4 minutes. Cooling the working fluid in ice
 280 cubes ensured full condensation of the collected boiler outflow. The mass of
 281 the collected fluid was measured via precision scale. The measured mass was
 282 then divided by duration of the test and the energy absorbed due to phase
 283 transformation of the fluid. In this way, the efficiency of the boiler for each
 284 power input was determined.

285 2.2.4. Pressurized Test

286 Following the Atmospheric test, the pressurized test was performed on
 287 the boiler. In this test, the boiler was operated at elevated pressure. This
 288 allowed investigation of capillary channel ability to effectively pump fluid at
 289 different operating pressures.

290 The test setup was similar to the setup presented in Figure 12, however,
291 this time making use of the clamp. The clamp provided obstruction to boiler
292 exit, allowing operation under a pressurized environment. Pressurized testing
293 commenced in similar fashion to atmospheric testing. First, the boiler was
294 heated to steady state at the given power input (1.8 W or 2.7 W). The
295 temperature and pressure data were recorded. Working fluid was introduced
296 based on the modeling effort described in Section 2.2.2 with initial supply
297 matching atmospheric test conditions. The boiler was then pressurized via
298 clamp application and the flow rate supplied to the boiler was reduced to
299 match the reduced mass flow leaving the boiler. This prevented an over-
300 supply condition to the boiler operation.

301 This technique was verified by visualizing the temperature and the pres-
302 sure profile of the boiler over the duration of operation. In these pressurized
303 tests, the test duration was 2 minutes. For each degree of restriction and
304 its adjusted flow rate, the temperature and the pressure data were recorded.
305 These tests provided the insight into saturation temperature and correspond-
306 ing pressure for the specific flow rate. However, to formally verify the op-
307 erating condition of the boiler during elevated pressure testing, additional
308 components and analysis were employed.

309 *2.2.5. Superheater Test*

310 Superheater components and tests were employed to fully validate and
311 characterize boiler operation. The use of the superheater was used in these

312 tests to fully validate boiler operation and working fluid thermodynamic
 313 state. The superheater successfully elevated the temperature of the working
 314 fluid beyond saturation conditions to full superheat as validated by $3M^{TM}$
 315 thermodynamic tables. By monitoring pressure and temperature of the work-
 316 ing fluid leaving the boiler and traversing the superheater, additional test
 317 points allowed characterization of boiler operation through the saturation
 318 conditions of the working fluid and along lines of constant pressure super-
 319 heat. In real-world applications, a superheater may be employed to increase
 320 energy available to connected power output devices.

321 Using the basic boiler test setup (Section 2.2), additional components for
 322 superheat of the working fluid were added that included pressure sensor, TCs,
 323 and two resistance heaters. First, the outlet from the boiler was connected
 324 directly to the in-house built superheater. Figure 13 shows the additional
 325 components employed along with arrows pointing data collecting locations.

Figure 13: Super heater test setup with all included components

326 In fabrication of the superheater, two micro-sized copper tubes of equal
 327 length were utilized. Each tube was carefully bent in serpentine manner to

328 allow maximum exposure to additional resistance heaters that were attached
 329 around the bends (see Figure 14(a)). This presented two distinct sections of
 330 the superheater where temperature and pressure data could be monitored be-
 331 tween sections. The heaters were then connected to separate voltage supplies
 332 with the capability of voltage and current read outs. T-tubing connected the
 333 two sections of serpentine superheater tubes and allowed an inline pressure
 334 sensor and TC to monitor working fluid conditions. A second T-tubing was
 335 connected to the outlet of the second superheater with a second pressure
 336 sensor and TC. Finally, boiler output was attached directly to one end of the
 337 combined superheater sections. Figure 14(b) shows the device with heaters
 338 and tubing enclosed below insulation marked SH1 (Superheat 1) and SH2
 339 (Superheat 2). All the connections were checked and then sealed properly to
 340 avoid any leakage of vapor during testing.

Figure 14: (a) Copper tube bent in serpentine manner, (b) Boiler test setup with connected superheater

341 Operation of the boiler and superheater commenced with the full setup
 342 heating to steady state. All temperature and the pressure data was recorded.
 343 Table 1 shows the full list of pressure recordings (by location) and tempera-

ture readings (by location). As before, working fluid flow rates were utilized based on atmospheric testing of the boiler, then reduced as operating pressure increased for each test. Several tests were conducted to investigate operating pressure versus saturated temperature and superheated temperature correlations. In this approach, first the vapor temperature was maintained constant in the entire setup (boiler and superheater). This allowed the operation at saturated region of the fluid. To operate at the superheated region, the power to the superheater was increased in steps to further raise the temperature of the incoming vapor from the boiler. For each test, resulting vapor temperature and pressure were noted. The test was then repeated for a different flow rate and operating pressure. The noted pressure data was then utilized to verify boiler operation along the isobars of the working fluid.

Table 1: List of pressure (P) recording and temperature (T) recording by locations

<i>Boiler (P,T)</i>	<i>SH1(P,T)</i>	<i>SH2 (P,T)</i>
Vapor temperature and pressure monitored inside the steam collecting dome or at the boiler exit	Temperature and pressure recorded at the outlet of SH1 or between the superheaters (SH1 and SH2)	Temperature and pressure of the steam recorded at the exit of the setup

355

2.3. Experimental Error

Kline and McClintock uncertainty analysis was utilized to evaluate the error associated with the test data [37]. The error associated with the voltage and the current readings were $\pm 0.1V$ and $\pm 0.001amp$, respectively. The error associated with each power input was $\pm 9mV$. Error associated with temperature, mass of the fluid and time were on the order of $\pm 0.5^{\circ}C$, $\pm 0.005g$ and

362 $\pm 1second$, respectively. The error associated with each individual evapora-
363 tion rate was determined to be in the order of $\pm 0.56 \frac{mg}{s}$. The error associated
364 with the pressure reading was $\pm 0.5kPa$.

365 **3. Result and Discussions**

366 The microboiler consisted of a boilerplate base and a steamdome covering
367 top. These parts were fabricated separately and then anodically bonded to
368 form the boiler. The fabricated boiler was then equipped with additional
369 components to fully ready it for testing. The tests were categorized as the Dry
370 test and the Wet test (Atmospheric and Pressurized). In the Dry test, the
371 operational characteristics were tested without supplying any working fluid
372 to the boiler. These included selection of the power inputs and monitoring
373 of the temperatures for the supplied power inputs at different locations of
374 the boilerplate. Using these temperatures, the computational model was
375 validated. The validated model was then utilized to discern heat available
376 at the channel for the evaporation of the working fluid. This generated heat
377 flux was further utilized to evaluate working fluid mass flow rate. These
378 combined results established the working baseline for the Wet test.

379 In the Wet Test, two sets of tests were performed on the microboiler
380 utilizing the established baseline from the Dry test. First was the atmospheric
381 test in which the boiler was operated in the atmospheric pressure. In this
382 test, the boiler ability to pump fluid out of the boiler via phase change was
383 evaluated. This provided one measure of boiler efficiency. Second, the boiler

384 was operated in a pressurized condition to investigate the working capability
385 of the capillary channels to draw fluid under the pressurized environment.
386 This test was then extended by including superheaters at the outlet of the
387 boiler. Results from this test were utilized to validate boiler operation along
388 the isobaric pressure line of the working fluid. Reporting of these test results
389 is presented in the following sections.

390 *3.1. Dry Test*

391 A resistance heater and the K-type thermocouples (TC) were utilized to
392 supply heat at the bottom of the boilerplate and to measure temperatures
393 at different locations of the microboiler, respectively. The power supplied
394 to the heater were selected to replicate real-world available heat sources as
395 noted in Section 2.2. Power inputs of 1.8 W and 2.7 W were selected which
396 yielded the heater temperatures of about 106°C and 144°C, respectively.

397 Power was supplied and the temperatures were recorded at locations in-
398 dicated in Figure 8. After steady state was achieved, the test was run for
399 approximately 15 minutes and the temperature readings were averaged (see
400 Figure 7). Table 2 shows the recorded temperatures. The average tempera-
401 ture at the boilerplate top were 105.5°C and 143°C for the power inputs of
402 1.8 W and 2.7 W, respectively.

403 *3.2. Thermal Modelling*

404 The Dry experimental test provided the power inputs (P_{in}) and corre-
405 sponding temperatures of the heater and the temperature at the top of the

Table 2: Temperature reading of the boilerplate

P_{in} (W)	Reservior (°C)	Center of Channel (°C)
1.8	142	144
2.7	105	106

boilerplate. This data was utilized in the model as described in Section 2.2.2. The designed model replicated real test conditions to accurately provide the heat flux available at the channel for the phase conversion of the fluid. The temperature distributions of the simulated model with 1.8 W power input is presented in Figure 15.

Figure 15: Temperature distribution with 1.8 W power supply

In order to verify the simulation, results obtained were compared to the Dry test temperatures. The comparison of the results are summarized in Table 3. The maximum discrepancy in the test and simulation model result was less than $\pm 5\%$. This validated the accuracy of the model.

The heat fluxes indicated by the model as available for phase change of the fluid were 1.14 W and 1.72 W for 1.8 W and 2.7 W, respectively. These

Table 3: Comparison of the Dry test and the model

Input Power (P_{in}) (W)		Average Temperature (°C)		Available Heat flux (W)
1.8	1.8	105.5	105.9	1.14
2.7	2.7	143	143.9	1.72

determined heat fluxes were then utilized to establish expected working fluid mass flow feed rates into the boiler for the Wet Test (Section 2.2.2.) The mass flow rate for each power input is presented in Table 4. The mass flow rates were $9 \frac{mg}{s}$ and $13.6 \frac{mg}{s}$ for the power inputs of 1.8 W and 2.7 W, respectively.

Table 4: Determination of the mass flow rate

P_{in} (W)	$P_{available}$ (W)	ΔH_v ($\frac{kJ}{kg}$)	$\dot{m}$ ($\frac{mg}{s}$)
1.8	1.14	125	9.12
2.7	1.72	125	13.6

3.3. Atmospheric Test

Following approximation of the mass flow rates, the boiler was ready for tests involving fluid with power input and fluid supply rates shown in Table 4. The typical temperature profile of the boiler before and after the fluid supply is illustrated in Figure 16 for power input of 1.8 W.

Vaporized fluid exiting out of the boiler was collected and weighed as described in Section 2.2.3. A set of three tests was performed for each power input. Results were averaged and are presented in Table 5.

The calculated averaged mass flow rate in Table 3 was combined with the heat of vaporization of HFE-7200 to determine energy absorbed via phase

Figure 16: Temperature profile of the heat source, the reservoir and the steam

Table 5: Measured temperature and the mass flow rate for each power input

P_{in} (W)	$P_{available}$ (W)	Source Temperature (°C)	Steamdome Temperature (°C)	Vapor (°C)	Mass Feed Rate ($\frac{mg}{s}$)	Average Mass Exit Rate ($\frac{mg}{s}$)
1.8	1.14	106	75.1	75.6	9	8
		106	75.1	75.6	9	
		106	75.1	75.6	9	
2.7	1.72	144	75.1	75.6	13.6	11.9
		144	75.1	75.6	13.6	
		144	75.1	75.6	13.6	

change of the working fluid and the efficiency of the microboiler. Figure 17 illustrates the comparison of the power input and the power observed via phase transformation of the fluid supplied.

The boiler demonstrated the phase conversion rates of $8 \frac{mg}{s}$ and $11.9 \frac{mg}{s}$ for the fluid inputs of $9 \frac{mg}{s}$ and $13.6 \frac{mg}{s}$, respectively. These values showed improvement versus boilers presented in prior efforts where a larger dimension acrylic steamdome was utilized [28, 30]. The use of a thin glass wafer

Figure 17: Supplied power vs power observed via phase change

Table 6: Efficiency of the microboiler

$P_{available}$ (W)	ΔH_v ($\frac{kJ}{kg}$)	$P_{evaporation}$ ($\frac{mg}{s}$)	Efficiency (η) (%)
1.14	125	1	87.8
1.72	125	1.48	86.2

as a steamdome eliminated a significant heat sink effect previously encountered. It should be noted that the temperature of the steamdome and vapor are almost equal. This allowed vapor to exit the boiler without premature condensation, thus improving the boiler operation.

Despite improved operation, some loss of working fluid mass was encountered. The boiler presented a maximum mass loss of about 12%. This loss was likely the result of the volatile nature of the HFE-7200 fluid combined with the capture and mass measure technique. To verify this assertion, an additional test was performed that replaced HFE-7200 fluid with water. This eliminated the volatile behavior of the HFE-7200 and verified the accuracy of the mass measurement.

449 As with HFE 7200 as the working fluid, the boiler was heated to steady
 450 state and water was supplied. The phased changed vapor was collected and
 451 condensed. The comparison of the measured mass flow rate out of the boiler
 452 at 1.8 W power input using water and HFE-7200 as the working fluid is sum-
 453 marized in Table 7. For comparison, results obtained using water demon-
 454 strated significantly reduced loss. Approximately 98% of the water supplied
 455 was collected following exit from the boiler. This is not an unexpected result
 456 as indicated by formal HFE properties supplied by 3MTM. The typical evap-
 457 orative losses for the HFE-7200 is approximately in the range of $339 \frac{mg}{(s-m^2)}$
 to $1017 \frac{mg}{(s-m^2)}$ when exposed to ambient temperature and pressure [38].

Table 7: Comparison of the mass flow rate for the different fluids

Workig Fluid Type	P_{in} (W)	Mass Feed Rate ($\frac{mg}{s}$)	Mass Exit Rate ($\frac{mg}{s}$)
Water	1.8	6	5.9
		6	
		6	
HFE-7200	1.8	9	8
		9	
		9	

458

459 3.4. Pressurized Test

460 The mass flow rate and power input were supplied to the boiler as shown
 461 in Table 4. The boiler was allowed to operate at atmospheric pressure and
 462 then the outlet tubing was restricted. The flow rate supplied to the mi-
 463 croboiler was then adjusted to match the mass flowing in and leaving out of
 464 the boiler as explained in Section 2.2.4. Figure 18 shows the temperature and
 465 pressure profile for each adjusted flow rate to the boiler with 1.8 W input.

Figure 18: Temperature (a) and pressure (b) trends by varying restriction to the boiler exit for 1.8 W power input

466 Similarly, Figure 19 shows the temperature and the pressure profile for
 467 each adjusted flow rate to the boiler for the 2.7 W power input.

Figure 19: Temperature (a) and pressure (b) trends by varying restriction to the boiler exit for 2.7 W power input.

468 The data in Figure 18 and Figure 19 were averaged and the standard de-
 469 viation associated with each temperature and pressure calculation are sum-
 470 marized in Table 8 for all power inputs.

471 Based on Table 8, pressure versus temperature data were plotted as a
 472 function of the mass flow rate for each power input (see Figure 20). The
 473 boiler operating temperature and pressure were increased as restriction on
 474 the boiler exit was increased. In good agreement with a saturated working

Table 8: The average temperature and pressure for each flow rate along with supplied power

P_{in} (W)	Mass feed rate ($\frac{mg}{s}$)	Vapor Temperature ($^{\circ}C$)	Pressure (kPa)
1.8	9	75.6	0g
	7	76.7 ± 0.14	3 ± 0.3
	5.5	77.6 ± 0.2	5.6 ± 0.2
	4.8	78.8 ± 0.2	8.8 ± 0.2
2.7	13.6	75.6	0g
	11.4	76.6 ± 0.3	3 ± 0.4
	10	78 ± 0.3	6.9 ± 0.4
	9.3	78.9 ± 0.35	8.5 ± 0.2

condition, pressure and temperature data demonstrated that vapor temperature followed the increasing trend of operating pressure inside the steam-dome. This was true for all power inputs as well as when these power inputs were compared to each other for particular pressure point. For the operating pressure of 3 kPa, the saturated temperature of the steam was approximately 76.7°C for both the power inputs. Such phenomenon were visualized for other operating pressures as well (see Figure 20). This validates for the specific saturated pressure there exists a unique saturated temperature and the boiler operation as saturated. Moreover, the pressurized test provided the saturated temperature and pressure points of the HFE-7200 working fluid.

To obtain the superheated properties of the HFE-7200 working fluid, the temperature of the steam exiting the boiler was further raised. The results from the test are presented in the following section.

Table 9: The temperature and the pressure recording of the vapor by location

P_{in} (W)	Feed Rate ($\frac{mg}{s}$)	Vapor Temperature (°C)			Operating Pressure (kPa)		
		Boiler Outlet	SH1 Outlet	SH2 Outlet	Boiler Outlet	SH1 Outlet	SH2 Outlet
1.8	7	76.8	76	76.5	3	3	3
		76.6	95	96.7	3	3	3
		76.7	105	107	3	3	3
1.8	5.5	77.5	76.5	77	5.6	5.6	5.6
		77.4	85	86.9	5.6	5.6	5.6
		77.4	101	103	5.6	5.6	5.6

Figure 20: The plot of boiler operating pressure versus vapor temperature for all power inputs

3.5. Superheater

Following the Pressurized tests, specific superheating tests were performed. This was achieved via inclusion of super heater as shown in Figure 14 on the output flow from the steamdome. Power of 1.8 W was supplied to the boiler and the flow rate and operating pressure were based on Table 8. The flow rates of $7 \frac{mg}{s}$ and $5.5 \frac{mg}{s}$ with the operating pressure of 3 kPa and 5.6 kPa were selected. The temperature and pressure data were recorded at different

locations of the setup (refer Table 1). Table 9 summarizes these recordings at the boiler and at the superheater.

The recorded data were then related to $3M^{TM}$ provided HFE-7200 thermodynamic tables to interpolate entropy at specific temperature-pressure points. Figure 21 show the plot of temperature versus entropy. The plot establishes the boiler operation along the isobaric line of the working fluid.

Figure 21: Temperature vs entropy plot of the boiler operation

With these insights into boiler characterization, further boiler development has been planned. The future work includes miniaturization of the superheater to incorporate it beneath the primary boiler plate. Such modification will allow the evaporated fluid exiting the steam dome to traverse the superheater utilizing a single heat source located beneath the boiler. This will better simulate a situation where a single heat source (like an Otto cycle) supplies heat to the entire system.

508 4. Conclusion

509 The efficiency of a thermodynamically limited process could be improved
510 by recovering the heat lost to the environment. One such approach is via
511 installation of a waste heat recovery system. The system recovers the heat
512 and converts it to useful work. Already established micro devices and engines
513 provide such opportunity at low capital cost. In this work, a MEMS based
514 boiler has been constructed and tested for such an application.

515 Several tests were performed to investigate the boiler operation. The Dry
516 test provided the selection of power inputs and temperatures. The selected
517 power inputs were 1.8 W and 2.7 W and the temperatures via conduction
518 were 106°C and 144°C, respectively. These temperatures were utilized to
519 verify the finite difference method (FDM) model of the boiler. The validated
520 model predicted the available power via conduction. The power was utilized
521 to calculate fluid feed rate for the Wet tests. A working fluid was selected
522 based on the realized operating temperatures of the dry boiler. In this work,
523 3MTM HFE-7200 was utilized. The evaluated mass flow rates were approx-
524 imately 9 mg/s and 13.6 mg/s for the 1.8 W and 2.7 W, respectively. These
525 flow rates served as the baseline fluid feeds to the boiler in Wet tests.

526 In Wet tests, the ability of the capillary channels to pump fluid under
527 atmospheric and pressurized environments was studied. In addition, a su-
528 perheated test was performed by raising the steam temperature exiting the
529 boiler. Based on available working fluid data for HFE-7200, a T-s chart was
530 plotted that graphically established key isobaric operation and thermody-

531 namic conditions of the boiler.

532 Operation of the boiler across these test points clearly showed the boiler
533 as a potential future waste heat scavenging device. Radial capillary channels
534 presented enhanced mass flow rate for phase transformation via capillary
535 flow. In addition, a 500 m glass steamdome inclusion in the design reduced
536 the boiler mass and the heat sink effect significantly versus prior boiler it-
537 erations. Such modification minimized the pre-condensation of vapor inside
538 the steamdome. A maximum efficiency of approximately 88% was demon-
539 strated by the boiler, an improvement of about 50% compared to previously
540 published work [28]. Furthermore, the operating pressure of the boiler ele-
541 vated as well. A peak operating pressure of 8.5 kPa was demonstrated by
542 the boiler.

543 With these significant improvements demonstrated by the present boiler
544 design, future work will be focused on incorporating a superheater beneath
545 the boiler. Such inclusion of a superheater will allow utilization of a single
546 waste heat source (like an Otto cycle) for the entire system. Careful de-
547 sign and material selection of the superheater will be considered in order to
548 maximize the heat transfer to the boiler via conduction.

549 **5. Acknowledgements**

550 *The authors gratefully acknowledge the support of this work by the NSF*
551 *via Grant No. ECCS-1053729.*

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