

Characterization and Optimization of a Radiantly Driven Multi-Stage Knudsen Compressor

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Abstract. The Knudsen Compressor is a micro/meso-scale gas roughing pump that operates by utilizing the thermal transpiration effect. A Finite Element Knudsen Compressor Performance Model was constructed and used to investigate the pumpdown performance of large-scale Knudsen Compressor cascades. Model predictions were compared to experimental measurements made for radiantly driven Knudsen Compressors of up to 15 stages. The temperature difference maintained across the thermal transpiration membrane was measured and agreed with analytically predicted values to within 15% except for gas pressures between 10mTorr and 1Torr, where the outward gas conduction cooling is transitioning to rarefied conditions and the 1D model is inadequate. A steady-state pressure difference of 120 Torr was achieved for a 15 stage Knudsen Compressor operating with an average pressure of 760 Torr of air and illuminated with 20.9 mw/cm² of radiant flux. Single stage optimization considerations are also discussed.

I. INTRODUCTION

The perceived utility of compact, power efficient sensor systems, coupled with recent advances in micro-electro-mechanical systems (MEMS) fabrication capabilities, have encouraged the construction of micro-scale and meso-scale sensors such as mass spectrometers¹, optical spectrometers², and gas chromatographs³; however, in many cases micro/meso-scale vacuum pumps, which are currently unavailable, are required to complete the sensor systems.⁴ The Knudsen Compressor, a micro/meso-scale roughing pump based on thermal transpiration, is one proposed design for a roughing pump for such applications. The thermal transpiration effect drives a gas flow and/or provides a pressure difference if a temperature gradient is maintained along the length of a channel in rarefied flow conditions. An apparently similar effect is commonly realized in membranes of porous materials such as aerogel. The Knudsen Compressor has no moving parts and requires no oil or supplementary fluids. Previous implementations of laboratory versions of Knudsen Compressors relied on resistive heating.^{5,6} The current work employs radiant in order to simplify manufacturing while at the same time minimizing the required power.

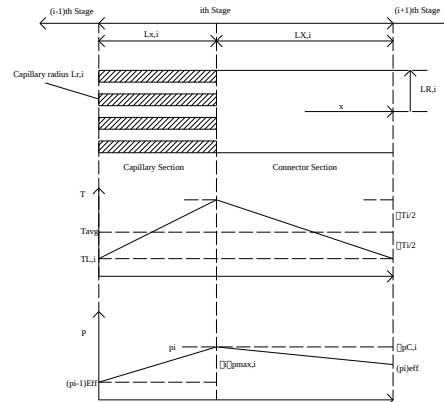


FIGURE 1. Illustrative i'th Stage of a Knudsen Compressor, Flow From Left to Right

Report Documentation Page				Form Approved OMB No. 0704-0188	
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1. REPORT DATE 13 JUL 2005		2. REPORT TYPE N/A		3. DATES COVERED -	
4. TITLE AND SUBTITLE Characterization and Optimization of a Radiantly Driven Multi-Stage Knudsen Compressor				5a. CONTRACT NUMBER	
				5b. GRANT NUMBER	
				5c. PROGRAM ELEMENT NUMBER	
6. AUTHOR(S)				5d. PROJECT NUMBER	
				5e. TASK NUMBER	
				5f. WORK UNIT NUMBER	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) University of Southern California, Dept. of Aerospace and Mechanical Engineering, Los Angeles, CA 90089-1191				8. PERFORMING ORGANIZATION REPORT NUMBER	
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)				10. SPONSOR/MONITOR'S ACRONYM(S)	
				11. SPONSOR/MONITOR'S REPORT NUMBER(S)	
12. DISTRIBUTION/AVAILABILITY STATEMENT Approved for public release, distribution unlimited					
13. SUPPLEMENTARY NOTES See also ADM001792, International Symposium on Rarefied Gas Dynamics (24th) Held in Monopoli (Bari), Italy on 10-16 July 2004. , The original document contains color images.					
14. ABSTRACT					
15. SUBJECT TERMS					
16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT UU	18. NUMBER OF PAGES 6	19a. NAME OF RESPONSIBLE PERSON
a. REPORT unclassified	b. ABSTRACT unclassified	c. THIS PAGE unclassified			

The Knudsen Compressor, first suggested by Pham-Van-Diep et al⁷ and demonstrated by Vargo et al^{5,6}, is a modern version of the original thermal transpiration compressor described by Knudsen^{8,9} in 1910. A series of individual stages are used to provide the required pressure ratio in a Knudsen Compressor. A single stage of a Knudsen Compressor is shown schematically in Figure 1, along with the temperature, (T) and pressure (p) variations along the stage. Each stage has a transpiration section, where the transpiration section is usually modeled by an array of capillary tubes for purposes of simplification.¹⁰ In this section the temperature increases, causing a pressure increase due to the rarefied gas dynamic phenomenon of thermal transpiration. The capillary membrane is followed by a connector section, with a significantly larger radius than the individual capillaries. The connector section operates with the gas flow closer to the continuum regime, so that the pressure is approximately constant while the temperature is lowered to its original value entering the stage. The compressor's potential applicability has been significantly expanded since Knudsen's 1910 investigations by the recent, serendipitous availability of small capillary membranes in materials such as aerogels, which have extremely low thermal conductivities.

II. KNUDSEN COMPRESSOR PERFORMANCE MODEL AND OPTIMIZATION

Silicon aerogel doped with a small amount of carbon (8%) was used as the transpiration membrane throughout this investigation. The pressure difference and net mass flow rate through a transpiration membrane are given by

$$\Delta P = p_{avg} \frac{\Delta T}{T_{avg}} \frac{Q_T}{Q_P} \quad \dot{M} = \frac{m}{kT_{avg}} C_{im} P_{avg} \frac{\Delta T}{T_{avg}} \frac{Q_T}{Q_P} \gamma (1 - \Delta) \quad (1)$$

Where Q_T and Q_P are the thermal gradient and pressure gradient driven flow coefficients^{11,12,13,14}, respectively. The parameter Δ ranges in value from 0 to 1 and represents the operating point on the pumpdown curve. Similar expressions can be written for the pressure difference and flow rates in the connector section, the effect of the connector section is neglected in the current study due to the low Knudsen Numbers ($Kn < .01$) experienced in the connector sections.

The temperature difference across the membrane is calculated using a 1D thermal model by balancing the radiant flux absorbed throughout the transpiration membrane and the cooling fluxes, the cooling includes; conduction through the membrane (solid conduction, radiation transport, and gas conduction), radiation outward, and outward free gas conduction and is described in detail elsewhere.¹⁵ The predicted temperature profile through the nominal aerogel transpiration membrane used in this study is shown in Figure 2 for various fluxes. The FE code, also described in detail elsewhere,¹⁵ kept track of the time dependent pressure variation within the pump allowing pumpdowns, vent ups, stage losses, and other events to be modeled.

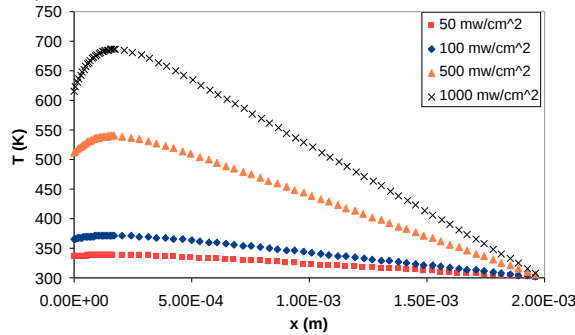


FIGURE 2. Temperature Profile Through the Nominal Aerogel Transpiration Membrane

The Knudsen Compressor Performance Model was also used to perform optimization studies. A Knudsen Compressor stage can be optimized under energy considerations by minimizing the energy per unit throughput and normalized pressure difference as shown by

$$\eta = \frac{\dot{Q}}{\dot{N} \cdot \frac{\Delta P}{P_{avg}}} = \left[\frac{L_x}{T_{avg}^2} \right] \cdot \left[\frac{1}{2L_r} \frac{Q_p}{Q_t} \right] \cdot \left[\frac{\sqrt{8km}}{P_{avg}} \right] \cdot \left[\frac{1}{(\gamma \Delta)^2} \right] \quad (2)$$

The first bracketed term represents how efficiently the radiant flux, $\dot{Q}$, can maintain a temperature difference across a transpiration membrane with a thickness, L_x . The second bracketed term represents the optimal scaling of the pore size with the operating pressure. These terms are plotted in Figure 3 for the case of 1 atm of N_2 . The area of the stages was varied to produce the same flow-rate of $6.05E17$ particles/s. The minimum of the first bracketed

term appears at a thickness of 0.5mm and a high flux of 1000 mw/cm². The minimum for the second bracketed term occurs at a Knudsen Number of roughly 1.15 for all of the pressures. The third bracketed term represents the operational conditions and is not involved in the optimization. The fourth bracketed term represents the operational location on the pumpdown curve and is minimized at a ϕ of 0.5.

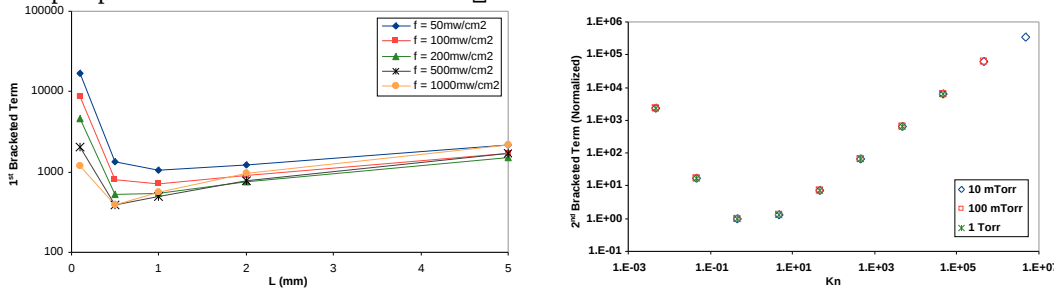


FIGURE 3. Minimization of First and Second Bracketed Term

III. DEVICE DESCRIPTION

The Knudsen Compressor cascades used in this study were fabricated, except for the transpiration membrane, using entirely conventional manufacturing techniques. They were designed according to manufacturing and experimental considerations and have not been fully optimized according to the analysis described above. A single stage radiantly driven Knudsen Compressor is shown disassembled in Figure 4 along with the corresponding 15 stage Knudsen Compressor.

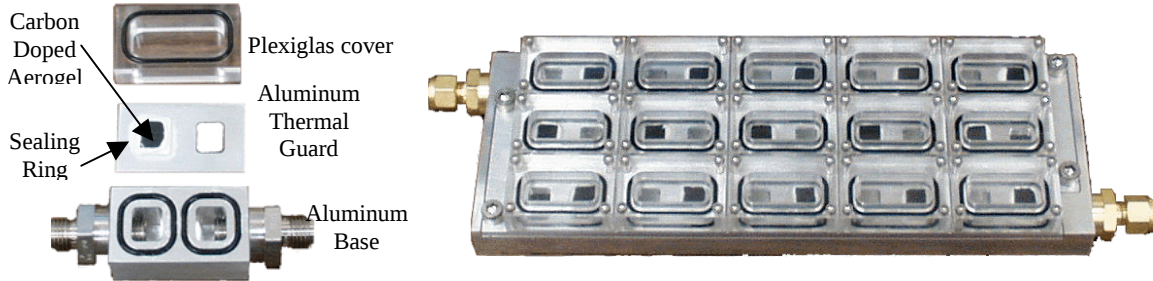


FIGURE 4. Radiantly Driven Single Stage Knudsen Compressor Parts and the Fifteen Stage Knudsen Compressor

The cascades are driven by illuminating the free side of an aerogel transpiration membrane through the Plexiglas[®] cover. Torr Seal[®] epoxy is used to seal the sides of the aerogel transpiration membrane and bond the membrane to the aluminum thermal guard, which is in thermal contact with the pump body. Silicon aerogel ($\rho=80\text{mg/cc}$) doped with a small amount of carbon (8%) was used as the transpiration membrane material. The Plexiglas[®] cover and aluminum base form the connector section of the pump. Different light sources, with different radiant fluxes, including a halogen lamp and Xenon lamps were used as the driving light sources for these measurements. Cascades of up to 5 stages were formed by connecting multiple stages together in series, while the 15 stage cascade had one aluminum base. Each stage of the 15 stage cascade was illuminated individually by a Xenon lamp.

IV. EXPERIMENTAL MEASUREMENTS

The transpiration membrane temperature difference was measured for various operating conditions. A single Knudsen Compressor stage was placed inside a 25cm diameter vacuum chamber. A halogen lamp with a radiant flux of 65mw/cm² at the location of the pump was used to drive the pump. The high temperature side of the stage was left open to the vacuum chamber (which had a much larger volume and therefore maintained a constant pressure during the run). The other side of the cascade was connected to a differential pressure sensor. The Plexiglas[®] cover was removed for the membrane temperature measurements to allow the thin film thermocouples to be placed in thermal contact with the hot side of the aerogel transpiration membrane. Pressure measurements were taken both

with the Plexiglas® cover in place and removed. The Plexiglas® cover was shown to reduce the radiant flux incident on the aerogel transpiration membrane by about 15%.

The temperature of the cold side thermal guard did not rise more than 4° during the experiments. Transpiration membrane temperature difference compared to the model predicted value is shown in Figure 5 for N₂.

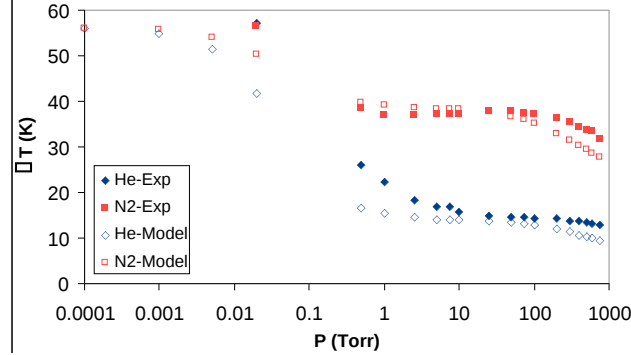


FIGURE 5. Transpiration Membrane Temperature Difference

There is a general agreement in functional form between the model predicted and the experimentally measured temperature difference for both gases. The downward trend for gas pressures between 100 Torr and 1000 Torr is due to the onset of gas conduction through the transpiration membrane. At high gas pressures the Knudsen Compressor Performance Model predicts a gas temperature that is lower than the measured value for both gases. In this pressure range the gas conduction in the pores of the aerogel is in the transitional regime and the outward cooling from the surface of the hot side is transitioning to convective thermal transport leading to uncertainty in the model. As the pressure is decreased from atmospheric the temperature difference for both gases increases due to decreased gas conduction through the transpiration membrane because of rarefaction effects. As this effect has become negligible compared to the other cooling mechanisms, at about 100 Torr, reducing the pressure further has very little effect on the temperature difference. At a pressure near 1 Torr the temperature difference begins to increase again due to a reduction in the outward gas conduction flux. This occurs experimentally, for both Nitrogen and Helium, at a higher pressure than predicted from the Knudsen Compressor Performance Model. The pressure range over which this transition occurs is also smaller for the experimental values as gas conduction outward cooling is completely negligible by a pressure of 20 mTorr, while the model requires about an order of magnitude lower pressure. Both of these transitional phases occur at slightly higher pressure for Helium because it has a smaller molecular diameter and hence a larger Knudsen Number for the same gas conditions. The exact location of the transition phases is not accurately defined, but the maximum temperature difference (for negligible gas conduction through the membrane and outward) and temperature difference that occurs with no gas conduction through the membrane are both predicted to within 8% for both Nitrogen and Helium.

The steady state pressure differences were measured using the same single stage Knudsen Compressor setup as for the temperature difference measurements. Figure 6 shows the maximum pressure difference produced at different vacuum chamber pressures of Helium and N₂ (with and without the Plexiglas® cover). Figure 6 also shows the variation in the pressure difference produced by the single stage with incident radiant flux for an atmospheric pressure of air.

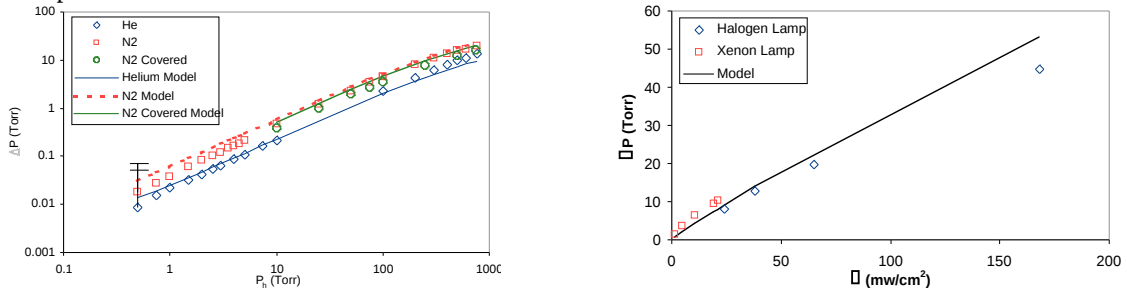


FIGURE 6. Steady-State Pressure Differences for Various Operating Conditions

The dominant trend of the pressure difference produced by the single stage Knudsen Compressor is a linear variation with the mean pressure of the stage, as anticipated by Equation 1. The decreased pressure difference of the covered version operating on N₂ corresponds to a reduction (roughly 15%) in the radiant flux striking the aerogel transpiration membrane due to the Plexiglas® cover. The pressure difference produced when operating on Helium is relatively smaller compared to N₂ because of the decreased temperature difference due to the increased gas

conduction through the membrane and outward. The agreement between the experimentally measured steady-state pressure differences and the values predicted by the Knudsen Compressor Performance Model is within the accuracy of the differential pressure sensor (50 mTorr) at low pressures as shown by the error bars included on the lowest pressure points in Figure 6 and is within 20% (roughly the discrepancy between the model predicted temperatures and the experimentally measured temperatures) over the intermediate pressures (up to several hundred Torr). At pressures above several hundred Torr the discrepancy grows corresponding to the growth in the ΔT discrepancy.

Two different light sources were used to provide the range of fluxes in Figure 6, a xenon lamp for $0 < \Phi < 25 \text{ mw/cm}^2$ and a halogen lamp for $25 < \Phi < 175 \text{ mw/cm}^2$. The xenon lamp provides a higher pressure difference for the same flux (as measured by the integrated photodetector). The different lamps have different spectrum and different angular flux distributions with the halogen lamp providing a broader distribution and the xenon lamp providing a distribution more similar to a point source which could lead to the discrepancy. At high temperature differences the model predicted and experimentally measured steady-state pressure differences deviate further. This is likely due to free convection cooling, which is currently neglected, becoming relatively more important at higher incident radiant fluxes. The model does predict a steady-state pressure difference that is between the experimentally measured value for the two different lamps and is adequate to radiant fluxes of roughly 100 mw/cm^2 .

Pumping curve measurements were taken for Knudsen Compressor cascades composed of 1, 2 and 5 stages. The Knudsen Compressors were connected between two known volumes and the pressure rise verses time was measured until the maximum pressure difference was established. Figure 7 shows the steady-state pressure differences for cascades with 1, 2, and 5 stages for various pressures of N_2 .

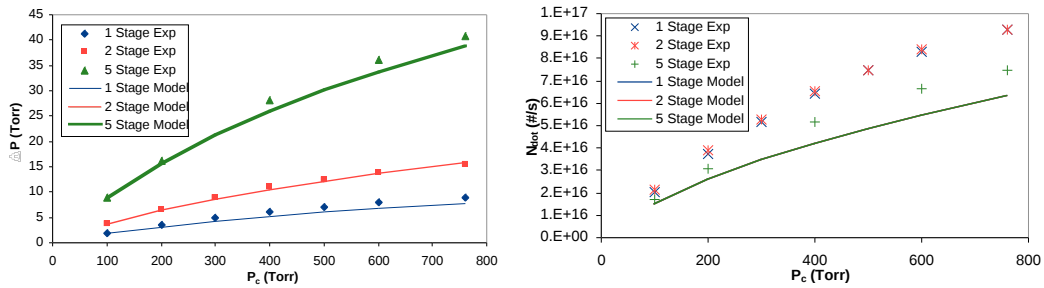


FIGURE 7. Pressure Difference and Throughput Produced by Various Numbers of Stages

The stages were individually illuminated by xenon lamps with a flux of 20.9 mw/cm^2 at the hot side of the aerogel transpiration membrane. Figure 7 also shows the steady-state pressure differences predicted by the Knudsen Compressor Performance Model. The agreement between the predicted and measured values is within 8% over the entire pressure range. Each stage produced a steady state pressure difference of roughly 8 Torr at atmospheric pressure. The maximum pressure difference scales with the number of stages, which is the expected trend for cascades with small numbers of stages. Figure 7 also compares the experimentally measured and predicted maximum flowrate. The Knudsen Compressor Performance Model predicts the same flowrate for the different numbers of stages because of the small transpiration membrane temperature differences and again underpredicts the gas throughput. The difference between the experimentally measured values for different numbers of stages is likely due to stage-to-stage variations in manufacturing. Stage variations affect both the temperature difference and the gas conductance of the transpiration membrane. The stage pressure difference depends only on the temperature difference, while the stage throughput depends on both the temperature difference and the stage conductance.

Figure 8 shows the pumpdown trace obtained with the 15 stage cascade operating at a mean pressure of 1 atm of air for a transient heating and a steady-state heating case.

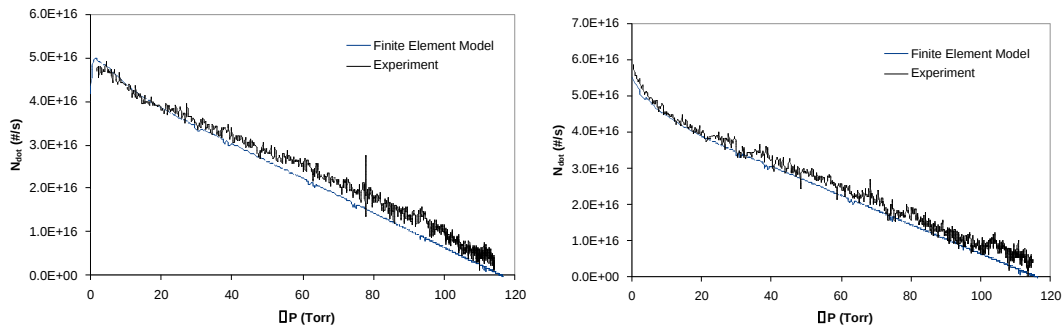


FIGURE 8. 15 Stage Cascade Pumpdown Curves

The 15 stage cascade produces a maximum pressure difference of 120 Torr or 8 Torr per stage. The 1,2 and 5 stage cascades also produced about 8 Torr per stage for the same conditions. The 15 stage cascade also produced a maximum gas throughput of 5×10^{16} #/s, which is lower than the 7.5×10^{16} #/s for the 5 stage cascade. Again this can be attributed primarily to manufacturing differences between the stages. The difference between the two experimental curves is a fundamental difference between transient heating startups and steady-state heating startups. The nonlinear trend of both experimental curves for low pressure differences is likely due to transient pressure gradients within the Knudsen Compressor immediately after startup.

V. CONCLUSIONS

Radiantly driven meso-scale Knudsen Compressor cascades ranging from 1 to 15 stage cascades were operated for a variety of gases, pressures, and radiant fluxes. A performance model was developed and used to predict the membrane temperature differences, maximum pressure differences, and maximum throughputs for these conditions. The experimental results and model predictions agreed to within roughly 15%. The Knudsen Compressor cascades tested here were not fully optimized according to the process identified in this paper. Attaining fully optimized cascades is the objective of continuing investigations.

ACKNOWLEDGMENTS

Our work on Knudsen Compressors is supported in part by NASA, award number NAG5-10399. The authors would also like to thank Dr. Steven Jones from the Jet Propulsion Laboratory for providing the aerogel samples.

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