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Vapor Compression Cycles for High Component Heat Loads on Next-Generation Small Satellites

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Abstract. High-power small satellites (SmallSats) have the potential to provide new and advanced capabilities; however, significant challenges prevent wide-spread use. Of these challenges, thermal management of high heat loads is most significant. This paper evaluates vapor compression cycles (VCCs) as a thermal management solution to increase the maximum component heat load at a fixed radiator area. The VCC with radiative heat transfer as a heat sink is described with analytical equations that couple the required compressor work input for a desired additional component heat load for a given second law efficiency. A figure of merit is defined. In a case study, the results of the equations are compared to a numerically modelled vapor compression cycle. Challenges for a realization of vapor compression cooling on-board SmallSats are pointed out.

1. Introduction

Small spacecraft handle increasingly complex tasks that include communication, scientific observation, remote sensing, surveillance, and data-relay services [1]. They can be roughly categorized as CubeSats and SmallSats, where a CubeSat is a special class of SmallSat built to a standard unit (or U) dimension of 10 cm x 10 cm x 10 cm. They typically range in size from 1U up to 27U, and weigh less than 1.33 kg (3 lbs) per U. Both CubeSats and SmallSats are increasingly complex, and the applications demand an increasing power density. Conventional satellites often rely on surface properties, attitude and survival heaters to stay within an acceptable temperature band, both when being exposed to sunlight and when being in the shade of earth. This design is now challenged by a satellite industry that would like to add even more electronic components to the satellite. Deployable radiators can be used to dissipate larger amounts of heat; however, the surface area is often limited by practical considerations. Increasing radiation emission by operating the satellite at a higher temperature is often impossible because the satellite components have maximum operating temperatures. It is then favorable to establish a temperature lift between the components and radiator surfaces, which suggests the idea of a vapor compression cycle.

Since the vapor compression cycle dominates terrestrial refrigeration and air-conditioning, the idea of utilizing it in space emerged almost naturally, as early as in the 1980s [2]. Since then, the vapor compression cycle (VCC) has been proposed for space applications as an active cooling system a number of times, as for example in [3], [4], [5] and [6]. The cycle shows favorable COPs and can be scaled for a range of capacities. Early studies quickly resulted in warnings about the oil-management in the compressor and the two-phase flow in zero gravity as a factor of uncertainty as pointed out for



example in [6] and [7]. A compressor that runs dry or a heat exchanger capacity far from the design point are significant concerns and may have been the reason for the very small number of VCCs that have experienced microgravity.

To the best knowledge of the authors, VCCs with microgravity flight experience in the English speaking literature are very few. There is the “Orbiter Refrigerator/Freezer” (OR/F) and its successor, the “Enhanced Orbiter Refrigerator/Freezer” [8], which was very similar in terms of volume, purpose and cold-side temperature ranges. During the same time period, another VCC was utilized and called the Life Sciences Laboratory Equipment (LSLE) [9]. The LSLE had a cooling capacity of approximately 100 W in refrigeration mode and with that, was bigger than the OR/F. The latest documented VCC in zero-gravity is the refrigerated centrifuge [10]. There appears to be no performance measurements or evaluation of failure modes available from studies involving these four VCCs. Apart from microgravity operation or testing, researchers have performed terrestrial system and component tests at varying orientations ([5] [11] [12] [13] [14] [15]).

Compared to the scarce microgravity system testing, two-phase flow has been investigated more thoroughly, both by inclination testing and in microgravity flight experiments. The efforts resulted in valuable criteria that allow a design engineer to apply equations for critical heat flux that have been provided for terrestrial usage also for designing systems in zero gravity [16].

This paper provides analytic equations that are meant to be a simple tool for SmallSat developers. The tool relates additional cooling capacity and compressor work for VCCs with a radiative heat sink. The equations use the second law efficiency to lump various cycle parameters into one parameter. The relationships are applied and interpreted in light of a simple 6U satellite in low Earth orbit (LEO) to give a more thorough understanding of the derived equations. This paper also aims at laying out important design considerations and open questions that come along with VCC development for SmallSats.

2. Thermodynamic SmallSat Model

2.1. Satellite model

The satellite without VCC is modeled to be at a single temperature (T_s) with one component heat load that is placed in LEO. The satellite temperature is assumed to be equal to the component temperature T_s . Figure 1 a) depicts this modeling approach. The satellite is shown as a sphere in the schematic but could be of any geometry since the following model is geometry independent. A typical satellite design process includes balancing the satellite temperature and the component heat load, given an environmental heat load calculated by designated software. For the satellite with one component heat load, the energy balance at steady-state is then:

$$\dot{Q}_e + \dot{Q}_l - \dot{Q}_r(T_s) = 0, \quad (1)$$

or

$$\dot{Q}_r(T_s) = \dot{Q}_e + \dot{Q}_l. \quad (2)$$

$\dot{Q}_e$ is the environmental heat load, $\dot{Q}_l$ the component heat load and $\dot{Q}_r$ is the rate at which the satellite radiates heat into space. For the case that a temperature lift is established by utilizing compression work $\dot{W}_c$, there are 3 temperatures (component, evaporator, and condenser) as compared with one temperature (T_s) for the baseline system model. Figure 1 b) shows that the satellite radiates at T_c (saturation temperature of the condenser in the VCC) and not at the source temperature (T_s). The component is cooled by contact with the VCC evaporator at T_e . The component temperature T_s is assumed to be the same as the baseline system and the satellite radiates at a higher temperature and therefore at a higher rate as compared to not utilizing a VCC. The higher heat rejection rate allows for additional component heat load $\Delta\dot{Q}_l$. The new radiation capacity is written as $\dot{Q}_{r,new}$ and the additional heat rejection is $\Delta\dot{Q}_r = \dot{Q}_{r,new} - \dot{Q}_r$. The model assumes an in-series heat flow of the incoming environmental heat load and the process of the VCC, meaning that the entire environmental heat load must also be pumped to a higher temperature level. The energy flows for utilization of a VCC are depicted in Figure 1, whereas the steady-state energy balance is given as:

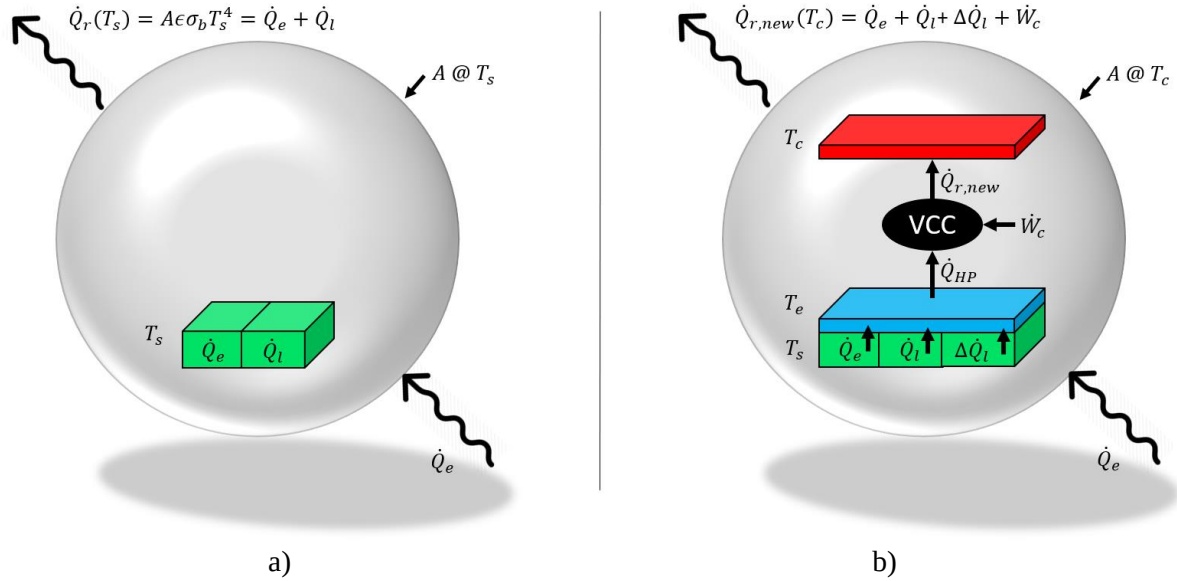


Figure 1: The modeled satellite with and without vapor compression cycle. In figure a), the component heat load $\dot{Q}_l$ and the environmental heat load $\dot{Q}_e$ are in an energy balance with $\dot{Q}_r$. In Figure b), it is attempted to utilize a VCC to allow for an additional heat load $\Delta\dot{Q}_l$ through a new radiation temperature $T_c > T_s$.

$$\dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l + \dot{W}_c - \dot{Q}_{r,new}(T_c) = 0. \quad (3)$$

The radiation temperature while using the VCC is set equal to the condensing temperature, denoted as T_c . The simple model development in the next sections assumes that the overall temperature lift for the VCC is $T_{lift} = T_c - T_s$. The effect of the temperature difference between the components and evaporating temperature ($T_s - T_e$) is embedded in a VCC second law efficiency.

2.2. Environmental model

In this paper, the described environmental heat loads and the component heat load are not distinguished but summed and expressed as the required heat rejection for the base case, $\dot{Q}_r(T_s)$, as determined using Equation (2) and (4).

2.3. Heat rejection in space

In space, heat can be rejected solely by radiation against the deep space temperature T_{DS} and the rate can be calculated using the Stefan-Boltzmann Law. The rejected heat $\dot{Q}_r$ when not using a VCC is shown in Equation (4).

$$\dot{Q}_r = A\epsilon\sigma_b(T_s^4 - T_{DS}^4) \quad (4)$$

The entire rejected heat $\dot{Q}_{r,new}$ when using a VCC is calculated equivalently by Equation (5).

$$\dot{Q}_{r,new} = A\epsilon\sigma_b(T_c^4 - T_{DS}^4) \quad (5)$$

The additional heat $\Delta\dot{Q}_r$ that can be rejected because of lifting the radiation temperature is then,

$$\Delta\dot{Q}_r = \dot{Q}_{r,new} - \dot{Q}_r = A\epsilon\sigma_b(T_c^4 - T_s^4). \quad (6)$$

The deep space temperature is set to 0 K for the following derivations.

2.4. On-board vapor compression cycle modelling

The basic four component VCC is analyzed by taking into account that the heat rejection is a function of the condensing temperature. This holds true when temperature gradients due to conduction through piping and conductive heat paths in the radiator are negligible.

In a first step, a figure of merit (FOM) is defined, starting with energy balances. Using Equation (3), the energy balance for the complete satellite with VCC is

$$\dot{Q}_{r,new}(T_c) = \dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l + \dot{W}_c, \quad (7)$$

or

$$\dot{Q}_r + \Delta\dot{Q}_r = \dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l + \dot{W}_c. \quad (8)$$

Assuming that the heat rejection at T_s is in balance with the environmental heat load and the component heat load as defined in Equation (1), it is:

$$\dot{Q}_r = \dot{Q}_e + \dot{Q}_l,$$

which reduces the overall energy balance of (8) to

$$\Delta\dot{Q}_r = \Delta\dot{Q}_l + \dot{W}_c,$$

or

$$\dot{W}_c = \Delta\dot{Q}_r - \Delta\dot{Q}_l. \quad (9)$$

Equation (9) indicates that the additional radiation capacity is distributed between the additional component heat load and the compressor power input. The ratio of the two is different from the commonly used COP. The compressor work needed to allow a certain amount of additional component heat load can be captured in a Figure of Merit, which is defined as

$$FOM = \frac{\Delta\dot{Q}_l}{\dot{W}_c} \neq \frac{\dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l}{\dot{W}_c} = COP.$$

To find an analytic expression for the FOM, the second law efficiency η_{2nd} is used to lump several efficiency penalties into one parameter. The penalties include throttling losses, compressor inefficiencies, and heat transfer over finite temperature differences. Two relationships shall be derived that describe the coupling of $\Delta\dot{Q}_l$, $\dot{W}_c$ and T_c for a VCC that uses radiation as a heat sink.

2.4.1. Work equation. To establish the temperature lift, the VCC needs to pump the environmental and component loads to a higher temperature level. For shorter notation, the total heat load is written as $\dot{Q}_{HP} = \dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l$. The second law efficiency η_{2nd} is defined and reformulated as:

$$\begin{aligned} COP &= \eta_{2nd} \cdot COP_{carnot} \\ \frac{\dot{Q}_{HP}}{\dot{W}_c} &= \eta_{2nd} \frac{T_s}{T_c - T_s} \\ \dot{W}_c &= \frac{\dot{Q}_{HP}}{\eta_{2nd}} \cdot \frac{(T_c - T_s)}{T_s}. \end{aligned} \quad (10)$$

The required condensation temperature can be expressed in terms of the new radiation capacity:

$$\begin{aligned} \dot{Q}_{r,new} &= A\epsilon\sigma_b T_c^4 \quad \text{since } T_{DS} \approx 0 \text{ K} \\ T_c^4 &= \frac{\dot{Q}_{r,new}}{A\epsilon\sigma_b}. \end{aligned} \quad (11)$$

Combining Equations (10) and (11) yields

$$\dot{W}_c = \frac{\dot{Q}_{HP}}{T_s \eta_{2nd}} \left(\sqrt[4]{\frac{\dot{Q}_{r,new}}{A\sigma_b\epsilon}} - T_s \right),$$

or

$$\dot{W}_c = \frac{\dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l}{T_s \eta_{2nd}} \left(\sqrt[4]{\frac{\dot{Q}_e + \dot{Q}_l + \Delta\dot{Q}_l + \dot{W}_c}{A\sigma_b\epsilon}} - T_s \right).$$

$\dot{Q}_e$ and $\dot{Q}_l$ can be written more compactly in terms of T_s (compare with Equations (2) and (4)) to find the Equation (12), which is written in terms of power per area:

$$\frac{\dot{W}_c}{A} = \frac{\sigma_b \epsilon T_s^4 + \frac{\Delta\dot{Q}_l}{A}}{T_s \eta_{2nd}} \left(\sqrt[4]{\frac{\sigma_b \epsilon T_s^4 + \frac{\Delta\dot{Q}_l}{A} + \frac{\dot{W}_c}{A}}{\sigma_b \epsilon}} - T_s \right) \quad (12)$$

Equation (12) is an implicit functional equation with $\dot{W}_c/A = f(\dot{W}_c/A, T_s, \Delta\dot{Q}_l/A, \eta_{2nd}, \epsilon)$. It is meant to be numerically solved for the compressor work $\dot{W}_c/A$ given a desired additional cooling capacity $\Delta\dot{Q}_l/A$ at a source temperature T_s that shall be maintained. This equation does not explicitly use or output the condensation temperature. Therefore, another equation is derived that couples the additional capacity with the condensation temperature.

2.4.2. Capacity equation. Combining Equation (10) and Equation (3) yields:

$$\dot{Q}_{r,new} - \dot{Q}_{HP} = \frac{\dot{Q}_{HP}}{\eta_{2nd}} \cdot \frac{T_c - T_s}{T_s},$$

or

$$\begin{aligned}\dot{Q}_{r,new} &= \dot{Q}_{HP} \left(1 + \frac{T_c - T_s}{\eta_{2nd} T_s} \right) \\ \dot{Q}_{HP} &= \frac{\dot{Q}_{r,new}}{1 + \frac{T_c - T_s}{\eta_{2nd} T_s}}.\end{aligned}$$

$\dot{Q}_{r,new}$ can be replaced by Equation (5) and by setting $T_{DS} = 0$ K to find:

$$\dot{Q}_{HP} = \frac{A\epsilon\sigma_b T_c^4}{1 + \frac{T_c - T_s}{T_s \eta_{2nd}}} \quad (13)$$

To find the additional cooling capacity in relationship of the condensation temperature, $\dot{Q}_{HP}$ is split:

$$\Delta\dot{Q}_l = \frac{A\epsilon\sigma_b T_c^4}{1 + \frac{T_c - T_s}{T_s \eta_{2nd}}} - (\dot{Q}_e + \dot{Q}_l).$$

By writing $\dot{Q}_e$ and $\dot{Q}_l$ in terms of T_s , it is found that

$$\Delta\dot{Q}_l = \frac{A\epsilon\sigma_b T_c^4}{1 + \frac{T_c - T_s}{T_s \eta_{2nd}}} - A\sigma_b \epsilon T_s^4,$$

or

$$\frac{\Delta\dot{Q}_l}{A} = \sigma_b \epsilon \left(\frac{T_c^4}{1 + \frac{T_c - T_s}{T_s \eta_{2nd}}} - T_s^4 \right). \quad (14)$$

Equation (14) relates the cooling capacity per unit area to the condensing temperature for a given emissivity, heat source temperature and second law efficiency. This equation can be used to quickly predict benefits of increased radiator temperature. The compressor work is then calculated using equation (12).

2.5. Summary

Equations (12) and (14) describe relationships for $\Delta\dot{Q}_l$ and $\dot{W}_c$ in terms of T_c and T_s for a VCC with a radiator for heat rejection, where T_s is a balance heat rejection temperature that meets the load determined as the sum of $\dot{Q}_e$ and $\dot{Q}_l$ and also acts as the source temperature of the heat pump. Table 1 lists both optimistic and conservative assumptions that have been used in deriving the work and capacity equations.

3. Model application and Discussion

3.1. Parametric study of the additional component heat load $\Delta\dot{Q}_l$

Using the *work* and *capacity equations* (Equations (12) and (14)), it is possible to plot the required compressor power, condensation temperature, Figure of Merit, and COP for varying additional heat loads. Figure 2 shows the results for second law efficiencies of $\eta_{2nd} = 0.3$ and $\eta_{2nd} = 0.5$. The

Table 1: Assumptions in deriving the work and capacity equation.

Conservative Assumptions	Optimistic assumptions
- The heat rejection is modeled with the condensation temperature although a portion of the heat is rejected at significantly higher temperatures due to superheated refrigerant exiting from the compressor discharge port. - In a real satellite, not all of the environmental heat load would have to go through the heat pump. Depending on the conduction paths, a portion of it would directly increase the radiation temperature.	- No temperature gradient has been modeled in the condenser. In any real heat exchanger, the effective radiation temperature would be below the refrigerant temperature. - The entire surface area of the satellite is assumed to radiate at the condensing temperature. Achieving such a wide distribution is practically challenging.

maximum allowable component temperature and therefore heat pump source temperature is set to $T_s = 40^\circ\text{C}$ and the emissivity is set to $\epsilon = 0.8$. An increase in additional component heat load is accompanied by a strong increase in the condensation temperature as can be seen in Figure 2 b). It should be noted that not only the additional component heat load $\Delta\dot{Q}_l$ needs to be pumped through the vapor compression cycle but also the original heat load $(\dot{Q}_e + \dot{Q}_l)$. Hence, even a small increase in $\Delta\dot{Q}_l$ requires significantly higher heat rejection temperatures. The increasing condensation temperatures with increasing additional component loads decrease the COP as can be seen from Figure 2 c). The compressor work increases almost linearly with $\Delta\dot{Q}_l$, which yields close to constant values for the FOM (Figure 2 d)). The equations do not provide the compressor discharge temperature, which may be a limiting parameter in the final design and can only be accessed by modelling a specific refrigerant.

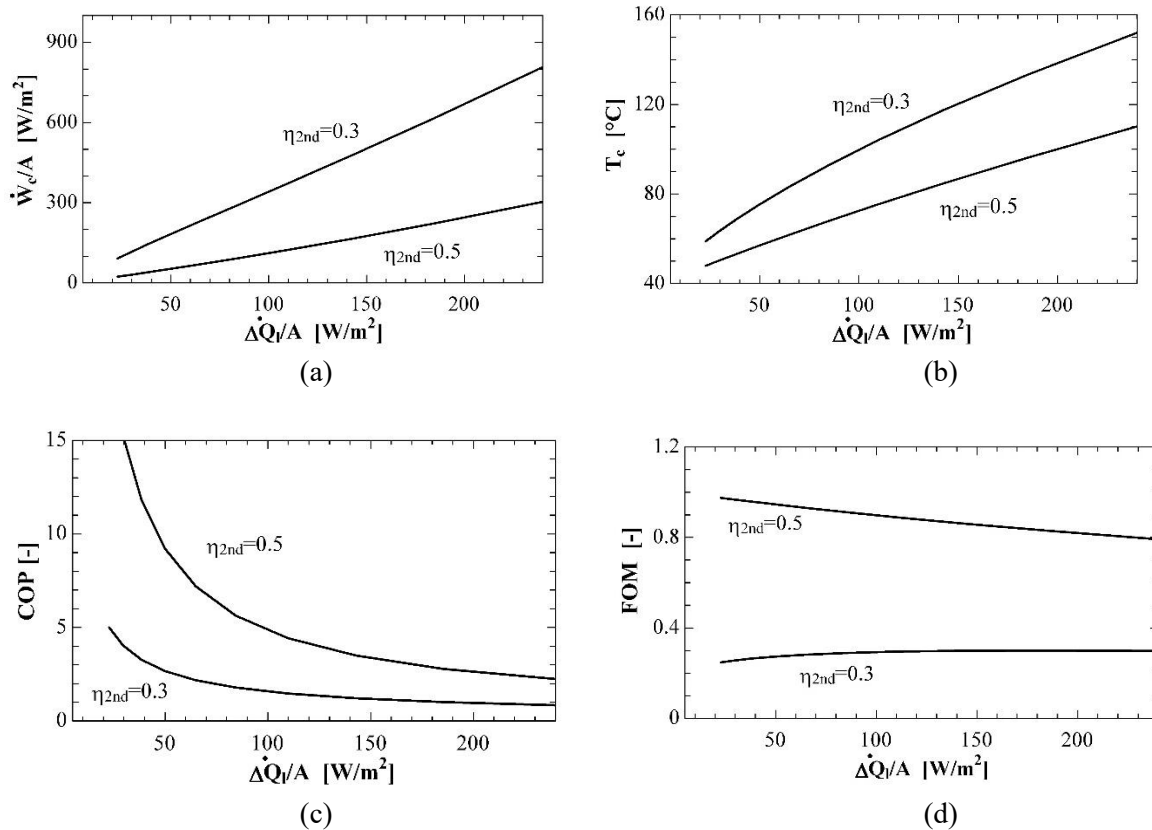


Figure 2: Area specific compressor work (a), condensation temperature (b), COP (c) and FOM (d) as a function of $\Delta\dot{Q}_l/A$ for $\eta_{2nd} = 0.3$ and $\eta_{2nd} = 0.5$ using Equation (12) and (14)

3.2. Model application to a 6U CubeSat for a hot case scenario

The model was applied to a spherical satellite that has an exterior surface area of $A = 0.22 \text{ m}^2$ consistent with that of a 6U CubeSat. The exterior optical properties are 0.2 (absorptivity) and 0.8 (emissivity). For these parameters, an environmental heat load of $\dot{Q}_e = 33 \text{ W}$ was found by using Thermal Desktop. Furthermore, a component heat load of $\dot{Q}_l = 63 \text{ W}$ resulted in a satellite temperature of $T_s = 40$. In order to evaluate the validity of the *Work Equation* (12) with a constant second law efficiency, its results were plotted against a thermodynamic model of a VCC with ammonia as the refrigerant. The model was used to determine the second law efficiency as a function of the additional component heat loads $\Delta\dot{Q}_l/A$. Table 2 shows the parameters for this simulation. The model was solved with the Engineering Equation Solver (EES). The compressor was modelled with a constant isentropic efficiency and the mass flow rate was determined to meet the heat load, simulating a variable speed compressor. A parametric study was conducted for varying additional component heat loads $\Delta\dot{Q}_l/A$. Figure 3 a) shows the results of the numerical model and of the work equation evaluated for three different second law efficiencies. The second law efficiencies are chosen to be 0.3 and 0.5 as in the previous section as well as 0.43, which is the maximum second law efficiency achieved in the numerical model. Figure 3 a) shows that the work equation and the numerical computation yield the same results if the second law efficiency is equal (i. e. $\Delta\dot{Q}_l = 112 \text{ W}$, $\eta_{2nd} = 0.43$). For $\Delta\dot{Q}_l$ values that are higher and lower, the second law efficiency decreases and so the prediction of the work equation with $\eta_{2nd} = 0.43$ diverges from the numerical results. If a satellite designer does not know the maximum second law efficiency, results can be approximated with more conservative or more optimistic assumptions for the second law efficiency like $\eta_{2nd} = 0.3$ and $\eta_{2nd} = 0.5$ as plotted. An additional heat load of $\Delta\dot{Q}_l/A = 68 \text{ W/m}^2$ was chosen to also evaluate different refrigerants (discussed in the next section).

Table 2: Parameters of cycle simulation

T_s	$T_s - T_e$	ΔT_{sub}	ΔT_{sup}	T_{DS}	$\dot{Q}_e + \dot{Q}_l$	A	$\eta_s [-]$	$\epsilon [-]$
40 °C	10 K	5 K	7 K	0 K	94.9 W	0.22 m ²	0.7	0.8
Refrigerant: Ammonia								

3.3. Design considerations and constraints

This section discusses the difficulties of implementation and summarizes some ideas on the feasibility of implementing a vapor compression cycle on a SmallSat.

3.3.1 FOM improvements: Figure 2 b) shows that the FOM is approximately 1 even for a high second law efficiency. This may not be enough to justify the installation of a VCC on a spacecraft. The FOM would be higher if only a share of the environmental and the component heat load had to go through the VCC. This could be achieved by thermally insulating the temperature sensitive component from the other heat loads. As a result, the heat loads that are not from the temperature sensitive components would dictate the radiation and therefore, the condensation temperature, but would not have to go through the VCC, which would reduce the work input. More detailed modelling is required to find the then achievable FOM.

3.3.2 Refrigerant: Figure 3 b) addresses the choice of the refrigerant. The bar plot was made for an operating point of $\Delta\dot{Q}_l/A = 68 \text{ W/m}^2$ or $\Delta\dot{Q}_l = 15 \text{ W}$. This is an appreciable additional cooling capacity and at the same time requires a compressor power that is achievable by a small scale compressor. For space applications, the question of toxicity and flammability may be answered differently than for terrestrial systems, hence, both flammable and toxic refrigerants are included. Satellite designers tend towards using flight proven materials, components and also fluids to lessen risks. Ammonia is used on the ISS in a single-phase loop through the radiators. Since it also shows a high second law efficiency for the evaluated 6U CubeSat, it was used here in the conducted case-study. However, the plot shows that Acetone results in even better performance. Also highly flammable

refrigerants should be considered. If no ignition source and no oxygen is present in the environment of the CubeSat, this may be a feasible choice. Moreover, in case of a leakage, the refrigerant would likely escape the CubeSat very quickly, since many CubeSats are not gas-tight.

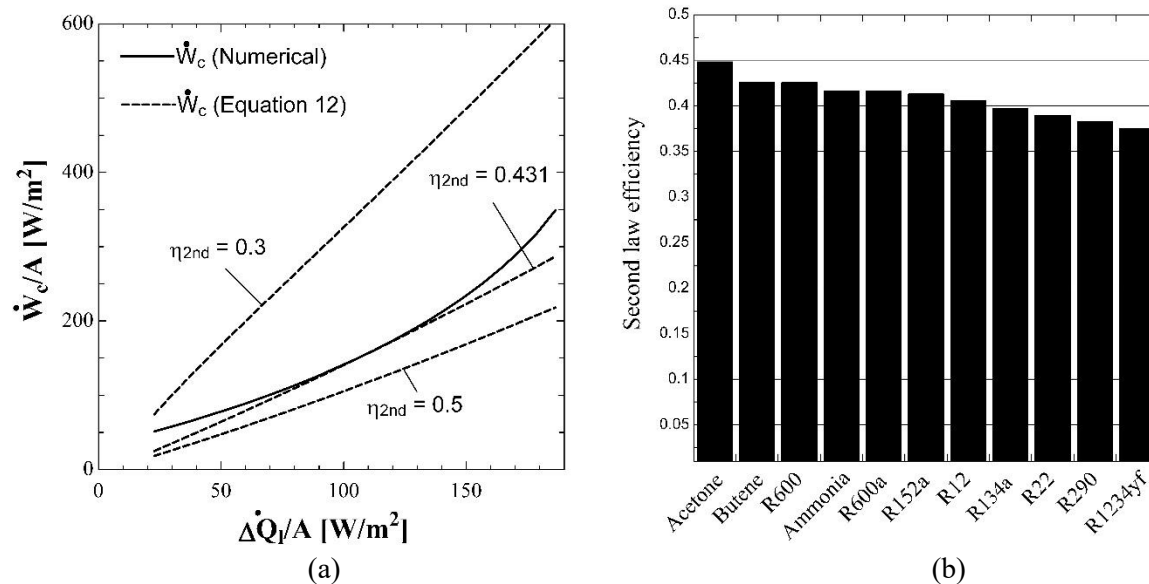


Figure 3: a) Compressor power as a function of additional component heat load. The solid line is from an EES model while the dashed line is calculated using Equation (12). b) Second law efficiency for an additional cooling capacity of 15 W for different refrigerants.

3.3.3. Heat exchangers: The conducted study assumed the entire area of the CubeSat radiator is at the condensation temperature. This would require that refrigerant flow through small channels along the outer shell of the CubeSat, spreading over as much surface area as possible. This may not be feasible and an actual design would result from trade-offs in cost and performance. On the evaporator design, it will be crucial to avoid exceeding the critical heat flux. A careful design with the expected payload heat loads in mind is necessary, potentially also using design guidelines for micro-gravity as discussed in [16].

3.3.4. Compressor: The compressor should be oil-free and small in volume as well as light in mass. To the knowledge of the authors, there is no off-the-shelf oil-free compressor that would suit the dimensions of a CubeSat. However, a powerful compressor that could serve a SmallSat has been discussed in [18]. The centrifugal compressor has a weight of 650 g and three in series were proven to provide 5 kW cooling capacity (1.63 kW each). Moreover, scroll type air-compressors can be oil-free and are available in volume and mass as small as 1.5 U and 500 g. Some development effort could result in an adaption to make this type of air-compressor a refrigerant compressor.

When designing compressor systems for spacecraft, the rotational momentums have to be considered, too. The startup and shutdown of the compressor induces a change in angular velocity, i.e. the satellite starts spinning around the axis of the compressor shaft. This effect can be counteracted by reaction wheels that are available as a typical CubeSat component. To avoid rotational momentum, linear compressors should be considered as well. They are known to work well even without oil and can be packaged very small, but would also induce vibrations.

3.3.5. Mass and volume: Mass and volume cannot be precisely predicted at this point. Nevertheless, a compressor of less than 1 kg, a condenser that is integrated in the CubeSat shell and therefore, barely causes additional weight and a highly efficient evaporator should be possible with less than 1.5 kg of

mass penalty. To make such a system interesting for 6U CubeSats, it should not take more volume than about 2U. For a SmallSat, more volume and mass could be allowed for the VCC.

3.3.6. Future work: The research on active cooling systems for SmallSats should be extended in different directions. A fair comparison with other cooling methods such as thermoelectric cooling or reversed Brayton cycle should be conducted. A prototype system should be built to prove that a reasonable size and weight can be achieved.

4. Conclusions

This paper considered energy implications for use of a VCC on a small satellite. Two equations were derived that allow simple estimates of the VCC work input ($\dot{W}_c$) and required condensing temperature (T_c) to achieve additional component heat load ($\Delta\dot{Q}_l$) in terms of the VCC second law efficiency η_{2nd} and other satellite specific parameters. A figure of merit was proposed as $FOM = \Delta\dot{Q}_l/\dot{W}_c$, which is generally below 1 even for high values of the second law efficiency. A thermal insulation of the temperature sensitive component would yield higher values for the FOM. Results were presented for a 6U CubeSat design. The work equation (Equation 12) matches the numerical results exactly when the second law efficiency in Equation 12 is equal to the second law efficiency found by the numerical model. Optimistic assumptions of the second law efficiency can be used to find conservative bounds of the VCC work input. The second law efficiency was also evaluated considering 11 different refrigerants for a constant additional component heat load of $\Delta\dot{Q}_l/A = 68 \text{ W/m}^2$. The second law efficiency ranges from $\eta_{2nd} = 0.38$ for R1234yf to $\eta_{2nd} = 0.45$ for Acetone.

5. Nomenclature

Symbol	Description	Symbol	Description
A	Area [m^2]	$\Delta\dot{Q}_r$	Additional rejected heat [W]
COP	Coefficient of performance $[-]$	T_e	Evaporation temperature [K]
COP_c	Carnot coefficient of performance $[-]$	T_c	Condensation temperature [K]
F	View factor $[-]$	T_s	Source/component temperature [K]
$\dot{m}$	Refrigerant mass flow rate [g/s]	ΔT_c	Temperature difference between condenser and radiator [K]
$\dot{Q}_e$	Environmental heat load [W]	ΔT_{sub}	Imposed subcooling [K]
$\dot{Q}_l$	Component heat load [W]	ΔT_{sup}	Imposed superheat [K]
$\dot{Q}_r$	Total rejected heat (without VCC) [W]	T_{DS}	Deep space temperature [K]
$\dot{Q}_{r,new}$	Total rejected heat (with VCC) [W]	$\dot{W}_c$	Compressor power [W]
$\Delta\dot{Q}_l$	Additional heat load [W]		

Greek Symbols

ϵ	Emissivity $[-]$	η_s	Isentropic compressor eff. $[-]$
η_{2nd}	Second law efficiency (COP/COP_c) $[-]$	σ_b	Boltzman-constant [$\text{W}/(\text{m}^2\text{K}^4)$]

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