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ABSTRACT

Vapor compression cycles would have many applications in the space industry if it was not for the uncertainty imposed by microgravity environments on two-phase systems. A first step towards Zero-G for technologies involving fluid dynamics can be terrestrial testing at different orientations. For vapor compression cycles, there is very little literature describing this type of research. This paper describes the anomalies encountered during the pursuit of a continuous operation of a R134a vapor compression cycle while positioning it at fixed angles around one axis between 0 and 360°. Experimental data was collected on a dedicated test stand across two configurations, one using a flat-plate evaporator and the other configuration using a tube-in-tube evaporator. Liquid flooding of the suction line was observed for both configurations but also continuous operation throughout a complete loop for certain cycle conditions. Charge migration towards the evaporator when it was put at the bottom was calculated based on differing measurements of the two mass flow meters in the liquid and suction line.

1. INTRODUCTION

Vapor compression cycles (VCC) that experience a change of orientation with respect to gravity are a niche research topic. To the best knowledge of the authors, only three papers discuss system level orientation experiments. Two of these found a clear effect of the inclination angle, the third one reported almost orientation independent measurements. The three studies are reviewed in the following three paragraphs.

Grzyll and Cole (2000) modified a commercial compressor to enable operation in microgravity on the International Space Station (ISS) in a refrigerated centrifuge. The oil-less hermetic compressor was built for R404A and the system had a cooling capacity of around 800 W. After modifying the compressor, the duty cycle of the complete refrigeration cycle to meet a certain setpoint was measured in all six basic orientations for a runtime of one hour each. A remarkable range of the duty cycle from 37% to 100% was found, showing great sensitivity to orientation. It was reported that in a certain adverse orientation, a two-phase mixture instead of subcooled liquid reached the expansion valve, causing the decreased performance of the unit. The refrigerated centrifuge was eventually installed on the ISS but the cooling system failed soon after on orbit operation (NASA, 2019). The failure mode is unclear and was not necessarily related to the lack of subcooling during ground testing.

Domitrovic et al. (2003) built a 10 kW refrigeration system in two versions: LBU1, using tube-in-tube heat exchangers and LBU2, using flat plate heat exchangers. Both systems used a scroll compressor following the reasoning: "*After several initial screening tests, the scroll compressor was selected for its desirable design of lubricant paths in which the lubricant is mixed with the refrigerant that flows through the system*". While LBU2 was proven to perform better due to the flat-plate heat exchangers, the inclination testing was only reported for LBU1. The entire system was tested at angles from 0° to 315° around one axis. Domitrovic compared the steady-state performance at varying heat source temperatures and concluded: "*With respect to orientation, however, there is little change in performance; 90° does* *appear to be the favored position, but there is never more than a 5% difference in either cooling capacity or cooling efficiency between any orientation at a particular EWIT.* [evaporator water inlet temperature]”. It remains unclear whether the flat-plate heat exchangers of LBU2 rendered it less stable with respect to the inclination angle. It is also unclear whether the system orientation was changed during operation (as opposed to the system being turned off) and if so, whether significant temporary deviations from the steady-state performance were observed as the angle was changed.

Sunada et al. (2008) tested a modified VCC using R134a with a cooling capacity of approximately 200 W. A flash tank was installed after the expansion valve and a gear pump fed the liquid phase into the evaporator. A proprietary wick in the evaporator should ensure 100% vapor going to the compressor and excess liquid being returned to the flash-tank. Although the inclination angle was changed only in the range of 15°, liquid flooding in the suction line correlated with the inclination angle.

The current paper presents anomalies identified in the pursuit of the continuous operation of a VCC through 360° with stable superheat and subcooling. The ultimate research goals include increasing the reliability of vapor compression systems for microgravity applications and preparing for successful microgravity experiments on parabolic flights. An interest in VCC technology for microgravity environments has existed for several decades. For example, Wilson (1988) worked on appliances for a space station (refrigerator/freezer, dishwasher and cloth drier) and noted with respect to the refrigerator/freezer: “*Certainly with enough project time and funding these problems* [maintain liquid seal on capillary tube, heat exchanger sizing, compressor development] *and the many other problems associated with microgravity operation of a vapor compression system could be overcome.*” Despite the hopeful statement, VCCs are still extremely rare in microgravity as summarized in Brendel et al. (2020b). This is in contrast to many cryogenic cooling technologies that have been frequently used in space for many decades (Ross, 2006). Some of them, like a dilution refrigeration system also were tested at varying inclinations to prepare for the microgravity operation (Roach and Helvensteijn, 1999).

2. INCLINABLE TEST STAND

2.1 Cycle, components, measurements and data processing

The test subject for this study is a four-component vapor compression cycle using R134a in two different configurations as shown in Figure 1. The main change from the first to the second configuration was a substitution of two flat-plate heat exchangers (of which only one was used at a time) with a tube-in-tube heat exchanger. Apart from changing the evaporator type, a suction line mass flow meter was added, the suction line became longer and the two-phase line from the expansion valve to the evaporator became shorter. The first configuration is described in detail in Brendel et al. (2020a) such that the following describes only the second configuration with a helical coil tube-in-tube evaporator.

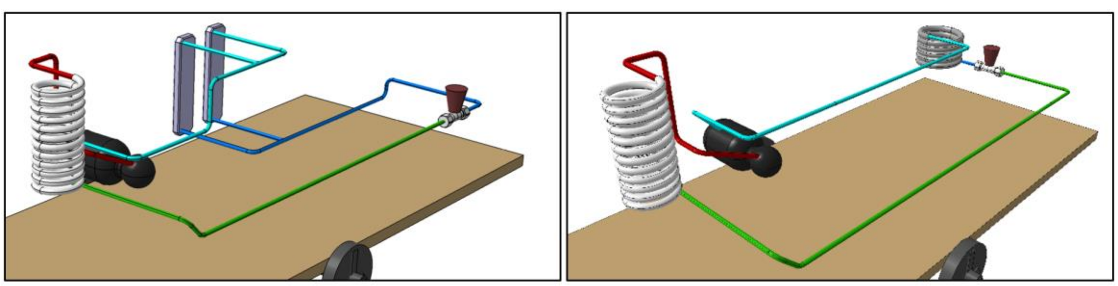

Figure 1: Refrigerant circuitry with flat-plate evaporator (left) versus with tube-in-tube evaporator(right).

Both configurations were tested with an oil-free commercial linear compressor and an oil-free prototype scroll compressor with water cooling of the housing and scroll pack. The controller of the linear compressor controls the piston stroke to draw power according to a setpoint manually given through a signal generator. The scroll compressor runs at a set speed manually adjusted through a voltage signal.

The expansion valve is a manually operated needle valve. Transparent tube sections have been installed in four locations of the refrigeration circuit. Figure 2 shows a schematic of the cycle with all components, sensors and the length of each tube section of the refrigeration circuit in centimeters. Data was recorded once every second and the raw data was processed with a five second moving average to flatten noise in the data. Post-processing involving fluid properties used the Python implementation of CoolProp (Bell et al., 2014).

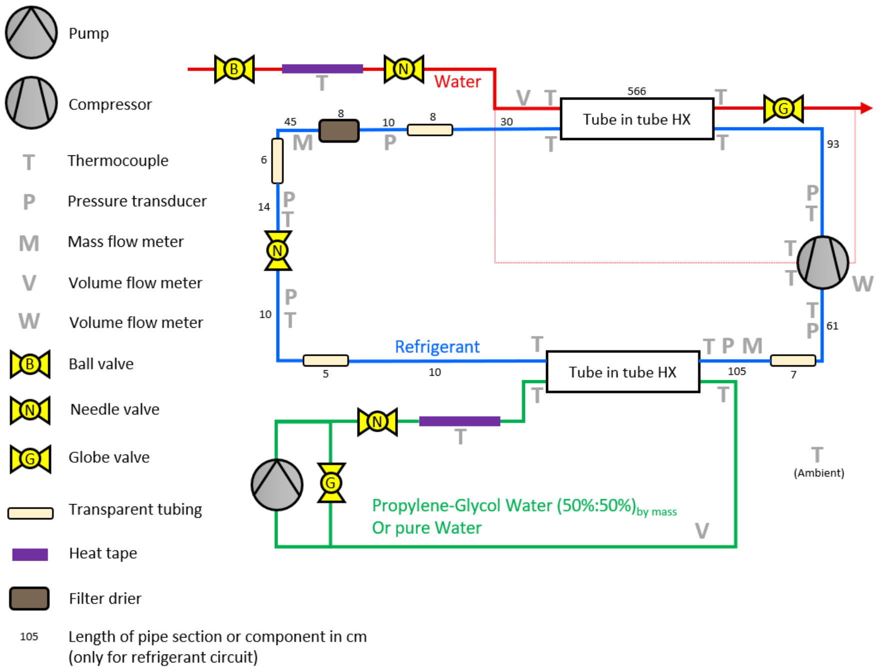

Figure 2: Schematic of tested cycle with tube-in-tube heat exchangers.

2.2 Circuitry of refrigerant and sign convention for inclination angle

The test stand was designed for operation in the horizontal position, which is the inclination angle $\theta = 0°$. The interpretation of some inclination test results requires knowledge of the relative component positioning. This is shown in Figure 1 and Figure 3. In the horizontal orientation, the compressor discharge enters the condenser at the highest point of the system from where it flows down in a helical coil condenser. The liquid line is horizontal and features a filter/drier and a Coriolis-type mass flow meter. After the expansion valve, the refrigerant enters the helical coil evaporator at the bottom and flows up. From the outlet of the evaporator, the refrigerant flows horizontally through a second Coriolis-type suction line mass flow meter and eventually flows down into the compressor.

The components are arranged such that inclination into one direction lowers both the evaporator and the expansion valve with respect to both the compressor and the condenser. This should support subcooled liquid at the expansion valve and superheated vapor at the compressor suction side and is defined as a positive inclination. Vice versa, if gravity pulls the liquid refrigerant away from the expansion valve or towards the compressor, the angle is defined as negative. This is exemplified in Figure 3. Subsequent experiments showed that the prediction of a positive angle is not always as trivial, but the definition will be kept for consistency.

2.3 Measurement uncertainty

This paper presents mostly results from temperature and mass flow meter measurements. The T-type thermocouples have a rated uncertainty of 1 Kelvin. However, verification showed that the measurements are within 0.4 K to a reference RTD thermometer both at a reference temperature of 0 °C and 40 °C. The uncertainty of the Coriolis-type mass flow meters is rated by the manufacturer to be 0.1 % of the reading. This uncertainty is negligible relative to the span of mass flow rates that is shown in the plots in this paper.

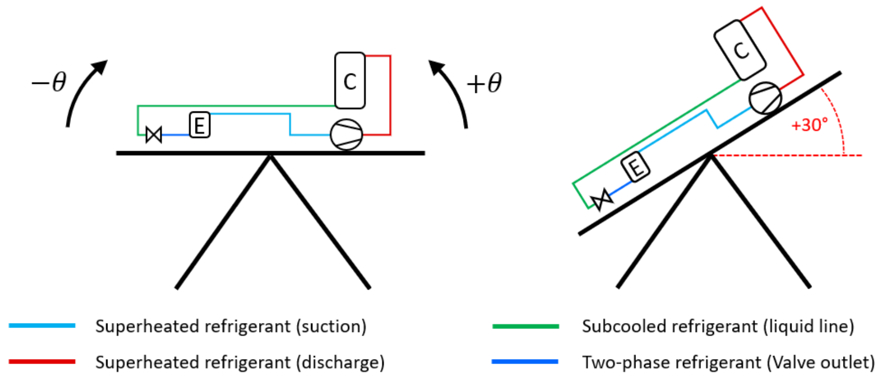

Figure 3: Definition of positive and negative inclination angles.

3. RESULTS

3.1 Orientation sensitivity of flat plate heat exchangers

Configuration: Flat-plate evaporators/linear compressor.

Situation: The cycle was at steady-state at an inclination angle of $\theta = -60°$. The heat transfer rate measured for the flat-plate evaporator was $\dot{Q}_{e,src} = 215\ W$ and $\dot{Q}_{e,ref} = 275\ W$ for the heat source and the refrigerant, respectively as shown in Figure 4. Note that of the two flat-plate heat exchangers shown in Figure 1, only one was used while the other one was closed with ball valves at the inlet and outlet. The heat source came into the evaporator with a temperature of $T_{e,src,in} = 13°C$ and was cooled to $T_{e,src,out} = -10°C$, superheating the refrigerant to $\Delta T_{sup} = 15\ K$, also shown in Figure 4. The error in the energy balance may be caused by the heat infiltration to the uninsulated flat-plate heat exchangers and the long connecting line from the expansion valve to the evaporators or the water-glycol flow rate being at the lower end of the flow meter range.

Change: The inclination of the test stand was changed from $\theta = -60°$ to $\theta = -90°$.

Observation: The inclination change caused the heat source heat transfer rate to drop to $\dot{Q}_{e,src} \approx 0\ W$ very quickly and the water glycol temperature at the evaporator outlet almost merged with the inlet temperature. At the same time, the suction superheat became 0 K, meaning that some amount of liquid was fed into the compressor shell and the refrigerant-side heat transfer rate cannot be accurately determined from the data. According to Figure 4, the heat source heat transfer changed before the inclination angle was changed. This was due to a problem in the data acquisition settings which gave the recording of the inclination angle a 20 second lag. The problem was solved later.

Hypothesis: The new inclination angle changed the flow pattern in the evaporator. At $\theta = -60°$, the refrigerant outlet was still notably higher than the inlet. The incoming liquid phase therefore formed a pool and spread over the entire width of the plates, allowing good heat exchange with the water-glycol brine. Gravity prevented liquid from reaching the evaporator outlet. At $\theta = -90°$, the refrigerant inlet and outlet to the heat exchanger were at the same height, such that the incoming liquid formed a stream at the bottom of the heat exchanger. This decreased the heat transfer surface area for boiling and increased the area of low effectiveness heat transfer from the water-glycol to the saturated or superheated vapor inside the heat exchanger. Figure *5* shows the cycle shortly after the measurements of Figure 4 at $\theta = -120°$. Frost accumulated only at the very bottom of the flat-plate heat exchanger which indicates where the boiling occurred. Frost accumulated also over the entire suction line and on the compressor shell which was not the case at an inclination angle of $\theta = -60°$. This is further evidence that the boiling in the heat exchanger was incomplete and occurred instead in the suction line and the compressor shell.

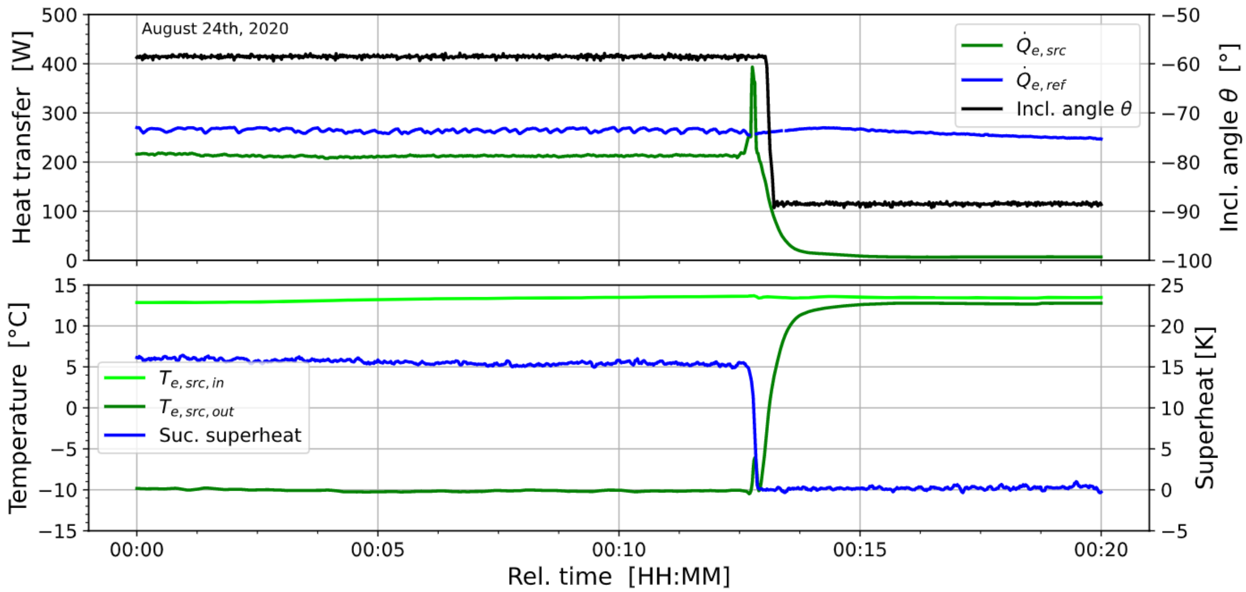

Figure 4: Loss of cooling capacity due to inclination of flat-plate evaporators.

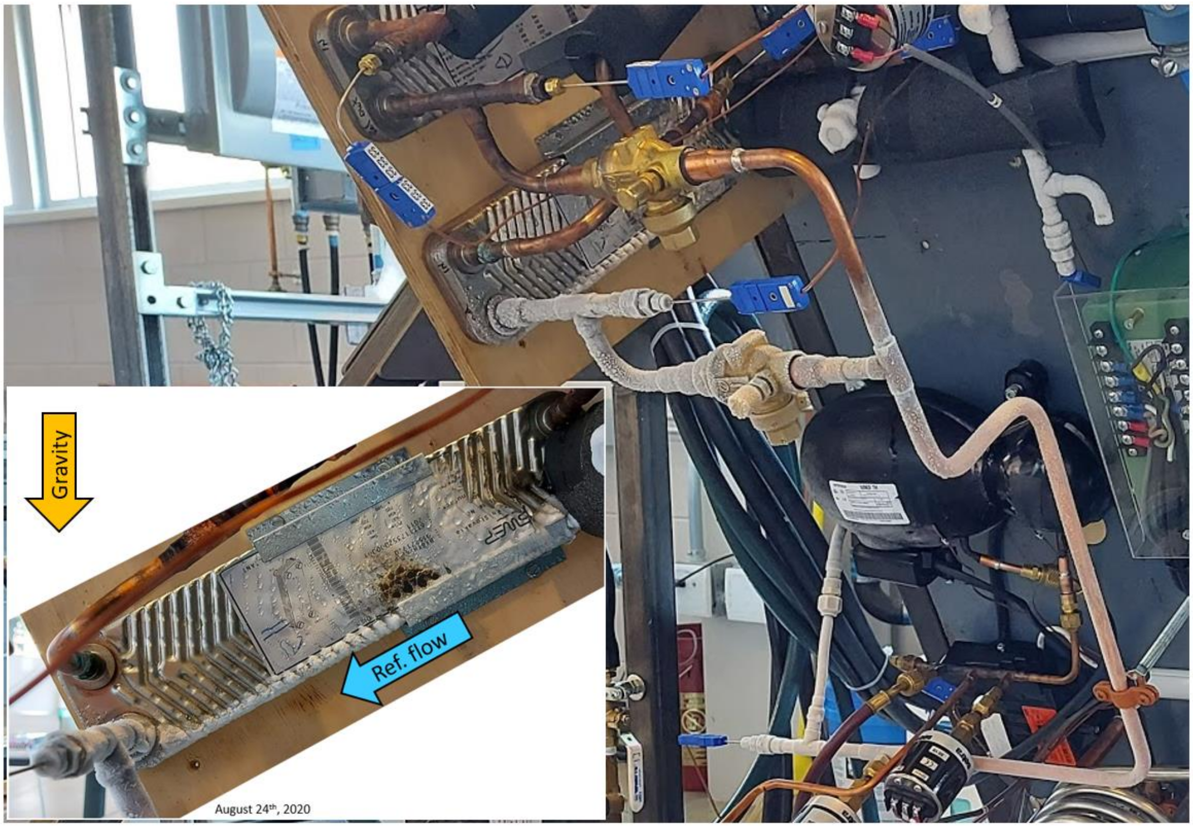

Figure 5: Incomplete boil-off in flat-plate heat exchanger with spatially separated frost accumulation along flow direction

3.2 Liquid flooding upon positive inclination angle

Configuration: Tube-in-tube evaporator/scroll compressor.

Situation: The test stand was in a horizontal position and the compressor suction temperature was $T_{suc} = 12°C$ and the saturation temperature corresponding to the suction pressure was $T_{suc,sat} = -10°C$. The mass flow rate was approximately $\dot{m}_{ref} = 2.3\ g/s$ measured by both mass flow meters (liquid line and suction line). Figure 6 shows this situation in the first 1.5 minutes of the plot.

Change: The inclination angle was changed to $\theta = +30°$, which lowers the evaporator with respect to the compressor. The suction line is then an inclined tube with upward flow, as can be seen from Figure 3. After 45 seconds, the test stand was turned back to the horizontal position ($\theta = 0°$).

Observation: The change in inclination angle caused an immediate loss of suction superheat. The measured suction line mass flow rate increased very quickly but the measurement was not reliable here due to the two-phase mixture in the flow meter. The liquid line mass flow meter detected a slow increase in mass flow rate followed by an equally slow decrease in mass flow rate after the stand was back in the horizontal position (Figure 6, right-hand side). The suction superheat recovered quickly. The observation is exactly opposite to the expectation of a negative inclination angle causing compressor flooding and a positive angle preventing it.

Hypothesis: The loss of superheat occurred very quickly considering that the suction line is more than 1.5 m long (see lengths in Figure 2). A possible explanation is that the inclination interrupted the stratified upward flow in the helical coil tube-in-tube evaporator. Instead, liquid plugs accumulated in one or several locations in the heat exchanger. Their repeated build-up and break-up threw liquid into the suction line which was carried by the vapor to the compressor. This would also explain the very unsteady measurements of the suction line mass flow meter.

In successive testing, similar changes in the inclination angle sometimes caused the same effects and sometimes not. A reliable correlation to operating conditions could not be verified. Empirically, an increased charge level seemed to increase the likelihood of the above observation.

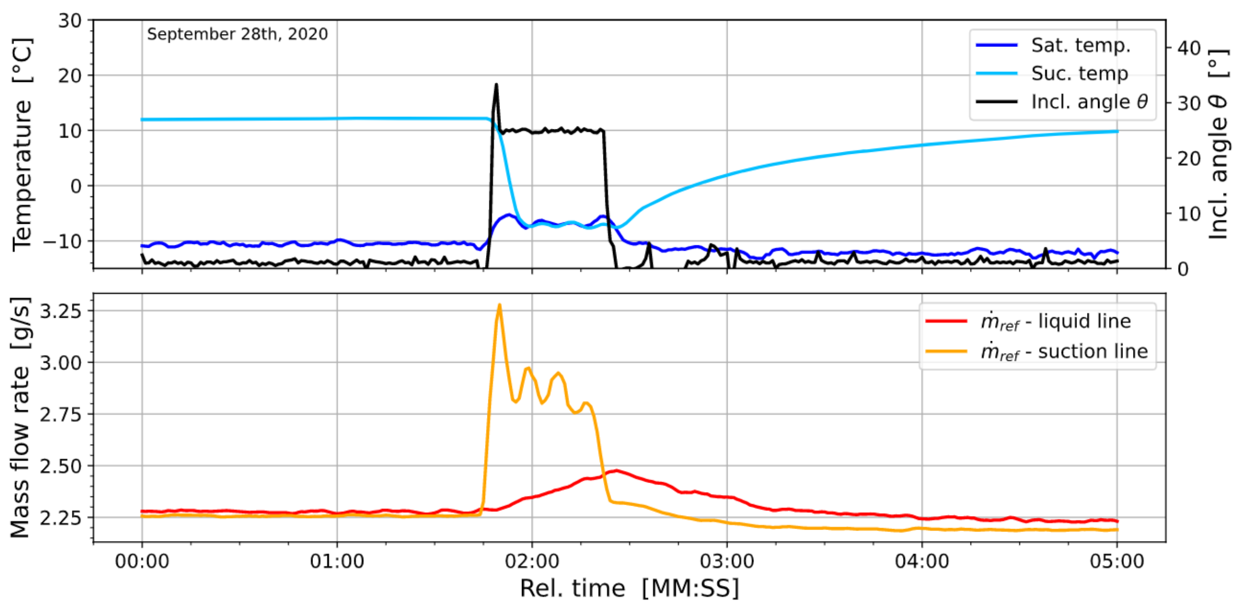

Figure 6: Liquid flooding of the suction line upon lowering the evaporator relative to the compressor.

3.3 Completed loop

Configuration: Tube-in-tube evaporator/linear compressor.

Situation: The cycle was operated at $\theta = 180°$ (up-side-down orientation). The mass flow rate was approximately $\dot{m}_{ref} = 1.4\ g/s$ with an evaporation temperature of $T_{e,ref,sat} = 8°C$ and a condensation temperature of $T_{c,ref,sat} = 26°C$. The superheat was $\Delta T_{sup} = 17K$ and the subcooling $\Delta T_{sub} = 5K$.

Change: The inclination angle was changed in increments of 10° from $\theta = 180°$ to $\theta = -210°$. Each angle was locked for one minute.

Observation: Figure 7 shows on the left-hand side a time plot of the inclination angle and the mass flow rate measured in the liquid line. As the inclination angle stepped through 390°, the mass flow rate varied between 1.20 g/s and 1.5 g/s. The right-hand side plot in Figure 7 shows the averaged data of 10-second samples of data, always recorded at the end of the one-minute time period of each 10° increment. The data is plotted with the inclination angle; hence it is chronologically read from right to left. The suction and expansion valve inlet temperature were very steady through all inclination angles. The strongest variations were visible in the evaporation temperature which varied between 0°C and 10°C and whose trend was well aligned with the mass flow rate.

Hypothesis: The particular operating conditions made the cycle *stable*, such that superheat and subcooling were maintained throughout all angles. The cycle was *not orientation independent*, since changes occurred that were clearly caused by the inclination angle. The change in mass flow rate is explained by the change in evaporation temperature and the associated changing suction gas density. The changes in the evaporation temperature are probably caused by a change of the heat transfer coefficients in the evaporator, but a detailed model is necessary to verify the possibility of this explanation. The change in condensation temperature correlates well with the mass flow rate and shows whether more or less heat had to be rejected.

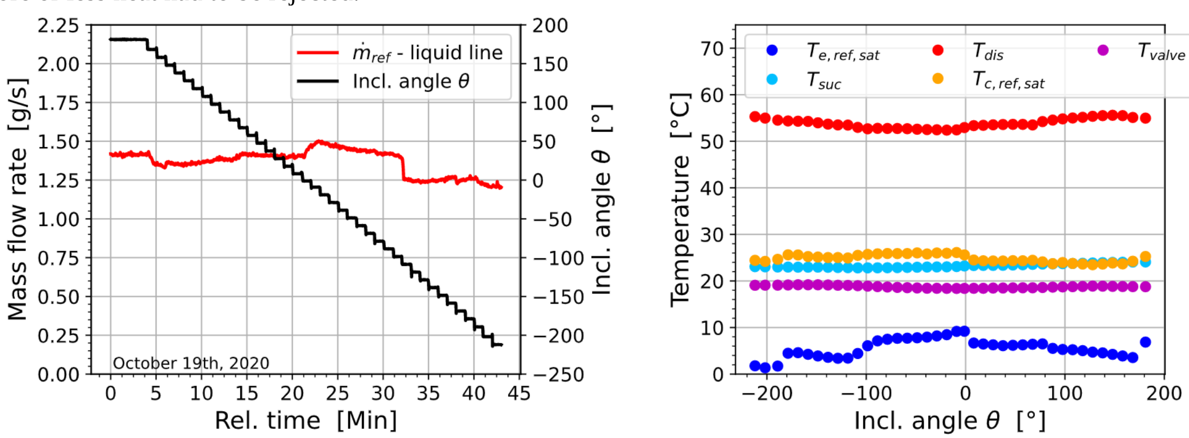

Figure 7: Stable operation of cycle through 390°.

3.4 Lasting change of mass flow rate due to change in inclination

Configuration: Tube-in-tube evaporator/scroll compressor.

Situation: The cycle was operated at steady-state with a mass flow rate of $\dot{m}_{ref} = 1.82\ g/s$ and an evaporation temperature of $T_{e,ref,sat} = 2°C$. The stand was in a horizontal orientation.

Change: The inclination angle was changed to $\theta = +90°, +180°, -180°, -90°$ for a period of 2 minutes at each angle. Then the stand was set back to a horizontal position. This is shown in Figure 8.

Observation: The suction line mass flow rate decreased with the first angle change and spiked high with the second. For both angle changes, the liquid line mass flow rate changed in the opposite direction but with a smaller magnitude.

At the end of the test, the stand was in a horizontal position just like at the beginning of the test, but the mass flow rate had decreased from 1.82 g/s by 11% to 1.62 g/s and seemed stable. The evaporation temperature had decreased by 3 K and showed an upward trend at the end of the test but only a very slow one compared to the changes that the different inclination angles had caused.

Hypothesis: The mass flow rate correlates well with the evaporation temperature and is a result of the changed suction gas density. To explain the lower evaporation temperature, it is useful to observe the charge migration which can be calculated using data from the two mass flow meters. For this purpose, they were "post-calibrated": At steady-state ($t = 0$), both mass flow meter readings were very close but deviated slightly ($\dot{m}_{ref,suc}(0) = 1.87 g/s,\ \dot{m}_{ref,liq}(0) = 1.83\ g/s$). The older suction line mass flow meter was therefore aligned with the newer liquid line mass flow meter using the difference at the initial steady-state condition with the following equation.

$$\dot{m}_{ref,suc,cal}(t) = \dot{m}_{ref,suc}(t) - (\dot{m}_{ref,suc}(0) - \dot{m}_{ref,liq}(0)) \tag{1}$$

If the liquid line mass flow meter reads more than the suction line mass flow meter after the calibration of the data, it means that the evaporator accumulates refrigerant and the condenser loses refrigerant, assuming that the single-phase sections of the circuit always hold the same amount of refrigerant. Hence, a charge migration can be defined as

$$\Delta m_m(t) = \int_0^t \dot{m}_{ref,liq} - \dot{m}_{ref,suc,cal}\, dt. \tag{2}$$

A positive charge migration means then filling the evaporator and a negative value is defined as accumulating charge in the condenser. The green line in Figure 8 shows that the charge migration rose during the +90° period but strongly fell thereafter. Ideally, the charge migration should be 0 when the stand is back in the initial orientation, but an accumulated charge migration of -10g was calculated. A smaller charge in the evaporator can explain the decreased evaporation temperature.

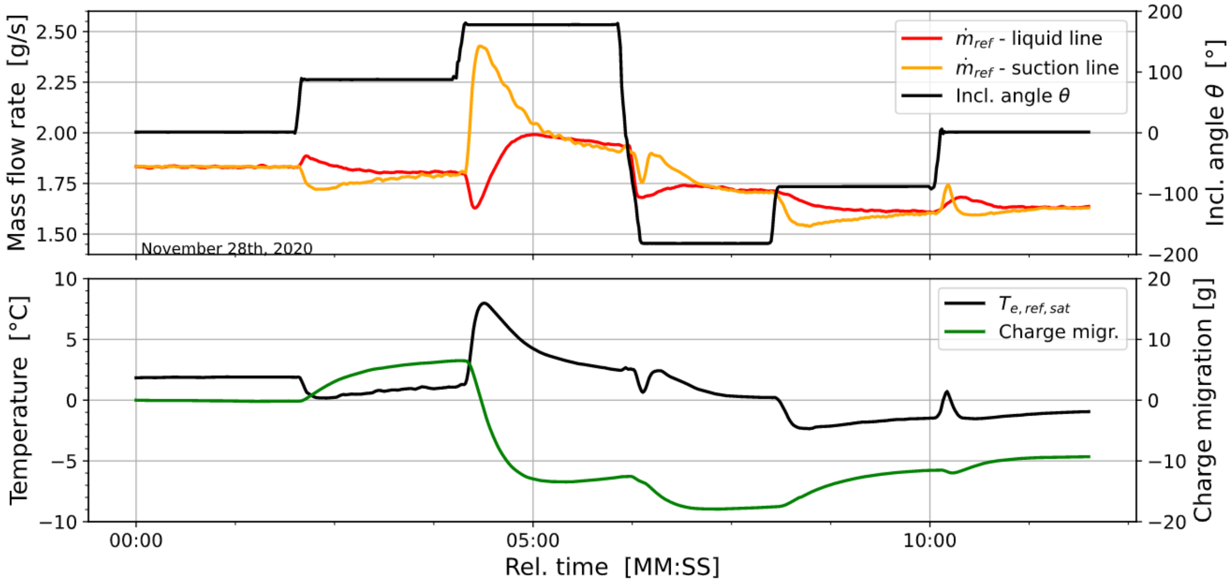

Figure 8: Marked shift in operating conditions after returning to a horizontal orientation.

Generally, the charge migration mostly follows intuition:

- $\theta = +90°$: The evaporator is at the bottom of the cycle and gravity acts against the low-pressure side flow direction, decreasing the suction line mass flow rate. The liquid line mass flow rate is barely impacted.
- $\theta = +180°$: The helical coil of the evaporator is a continuous downflow for the entering two-phase refrigerant. Hence, the refrigerant in the evaporator is "dumped out", causing a short large spike in mass flow rate. The refrigerant in the condenser has to flow upwards, causing a short decrease in the liquid line mass flow rate followed by realignment of the two mass flow rates.
- $\theta = -180°$: Although disturbed by the rotation, the trends started at $\theta = +180°$ continue.

It should be tested whether the operating conditions eventually realign with the initial ones. If not, this would mean that a given heat load, cooling water temperature, compressor speed, and charge do not uniquely define the refrigerant temperatures and mass flow rate. A charge sensitive model could also be used to study this ambiguity.

4. DISCUSSION

Only three papers were found in the open literature that discuss the effects of inclination on a vapor compression cycle performance. The field of research is wide open. The general importance as well as the most interesting research directions are unclear. Academically, the topic could benefit from a significant experimental campaign and the development of correlations that could be used for design analysis. However, the practical relevance is not yet well defined. An inclination independent cycle is not necessarily gravity independent. However, at least a better understanding of two-phase cycles exposed to zero-gravity is likely a likely practical outcome of continued research. Other applications of said research field could be in transport refrigeration.

A major untapped research question is how to define and correlate the stability of a VCC against inclination or gravity changes. A definition for "orientation independence" is virtually non-existent. It could be based on steady-state or transient measurements, changes in direct temperature measurements or the calculated cooling capacity, formulated for one axis of rotation or operation in six different orientations. Most two-phase flow research suggests that the cycle stability should increase with higher mass flow rates, but none of the three referenced papers on this topic attempted to validate this hypothesis.

This paper clearly showed orientation dependence of a flat-plate heat exchanger. It is unclear whether the orientation dependence can be diminished by an increased mass flow rate. Moreover, the heat exchanger has only been tested in orientations with the ports facing sideways. The results for other orientations cannot be inferred from the results presented and would require separate experiments.

The quick liquid flooding against gravity (liquid flowing upwards) was counterintuitive. A hypothesis has been presented in this paper but requires a transparent helical coil heat exchanger of similar geometry to be verified.

5. CONCLUSIONS

Experiments using a vapor compression cycle on an inclinable test stand showed that the performance of flat plate evaporators were very sensitive to their orientation. Using a tube-in-tube evaporator, a change in the inclination angle could sometimes lead to sudden liquid flooding in the suction line of the compressor against gravity despite more than 20 K superheat in the horizontal position. A continuous operation of a VCC over 390° with only small changes in the mass flow rate and temperatures was shown. The mass flow rate changed in a band of 0.25 g/s and the evaporation temperature changed in a band of 10 K. By comparison of the data from two mass flow meters, a charge migration was estimated and explained by the changing gravity vector. For the given operating parameters, more than 20 g of charge migrated in approximately 3 minutes. Overall, the research field has many unanswered questions but their relevance to industry is unclear.

NOMENCLATURE

Symbols and acronyms			**Subscripts**	
ISS	International Space Station		c	Condenser
$\dot{m}$	Mass flow rate		cal	Calculated
$\dot{Q}$	Heat transfer rate	g/s	e	Evaporator
T	Temperature	W	dis	Discharge
t	Time	°C	in	in, inlet
VCC	Vapor compression cycle		liq	Liquid
			m	Mass
Greek symbols			out	out, outlet

θ	Inclination angle	°	*ref*	Refrigerant
ΔT	Temperature difference	K	*sat*	Saturated
			src	heat source
			sub	Subcooling
			suc	Suction
			sup	Superheat
			valve	expansion valve (inlet)

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