AN ELECTRIC COOLING SYSTEM FOR AN UAV

Towards a bleed-less technology

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Abstract

Within this project the possibility to build a very light and yet effective cooling machine has been investigated. The purpose is primarily to cool electronics in an Unmanned Aerial Vehicle (UAV). Electrically driven cooling machines have been in focus of the project to suit the More Electric Aircraft (MEA) concept.

The normal method of cooling aircrafts today is by using a reversed Joule process, also called a reversed gas turbine process. The generally most used cooling machines, both ground-bound and mobile applications, are however all based on the vapor compression cycle, the typical electrical chiller cycle. The latter is more efficient, and thus when including the weight of the fuel needed to power the cooling cycle, it is very likely that the total weight of fuel and the cooling machine itself can be made lighter.

Components that could lower the weight of vapor compression machines have been studied, such as new lightweight heat exchangers, motors and compressors. Especially small lightweight compressors would be a key component to further enhance the competitiveness of the Rankine chiller cycles.

1. COOLING CYCLE COMPARISON

The Joule cycle, normally used in aircrafts, has a low efficiency due to the following reasons (see figure 1 and formula 1 below):

1. The jet engine efficiency driving the air to be compressed is lower than 30% normally.
2. The pressure at which air is bled off from the compressor is normally higher than necessary, resulting in a high temperature, T5.
3. The compressor efficiency is low which also increases T5.
4. The ram cooler experiences the stagnation temperature of the air outside resulting in a higher T6 and consequently a higher T7.
5. The expander efficiency is low which also increases T7.

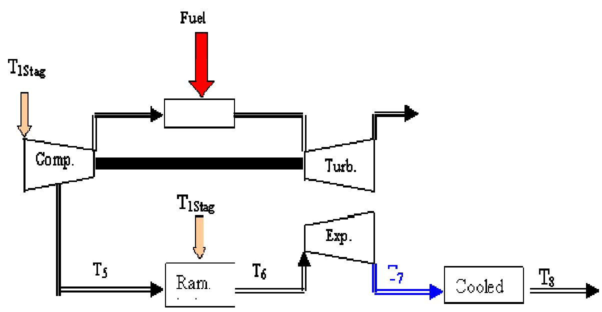

Fig. 1 A very simple reverse Joule cycle cooler for aircrafts

A very simplified formula of the reverse Joule cooling process can be written:

$$\eta_{Fuel\text{-}Cool} = \eta_{shaft} \frac{(T_8 - T_7)}{(T_5 - T_{1Stag})} \qquad (1)$$

$\eta_{Fuel\text{-}Cool}$ in this formula denotes the cooling power divided by the fuel power used. η_{shaft} denotes the engine overall mechanical efficiency. In table 1, a number of different reverse Joule coolers, used commercially, have been collected. Data are only indicative as many data from the coolers were not available – and were estimated.

Some conventional coolers using bleed air	Weight of cool-pack only kg	Fuel consump-tion kg/h	Cooling power kW	Specific power for 1h airtime kg/kW
Aircraft 1	10	9.4	7	2.77
Aircraft 1 updated	20	37.6	25	2.30
Aircraft 2	80	46.7	55	2.30
Aircraft 3	75	65.9	26	5.47
Average				3.21
Project goal	15	1.66	10	1.67

Table 1. Table 1. The efficiency and fuel need for four different air craft Joule coolers

The efficiency going from fuel power to cooling power is in average 7%. The cooling machine often uses fuel corresponding to 85% of its own weight/hour. Even if this is only valid during maximum cooling power situations, it is a strong indication that efficiency improvements are desired.

Attacking the points 1-5 in the beginning of this paragraph can surely improve the prevalent Joule cycle. It is however unlikely that, if an electrical drive motor is chosen, the reverse Joule cycle will retain its attractiveness as the preferred cooling cycle in aircrafts.

The direct vapor compression cycle can be built either in a direct fashion, cooling air, or in an indirect fashion, using secondary fluids for the heat transport.

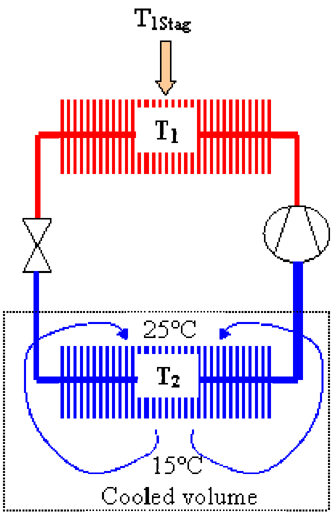

Fig. 2 A direct vapor compression cycle for cooling with cooled air.

The direct vapor compression cycle, fig 2, works between two main temperatures T1 in condensation and T_2 in evaporation. If cooled by ram air, T_1 must be somewhat larger than the stagnation temperature of the ram air T_{1stag} and T_2 must be somewhat lower than the outgoing air to the cooled volume. The stagnation temperature can be calculated as:

$$T_{1stag} = T_{air} + \frac{w^2}{2 \times C_p} \qquad (2)$$

where T_{air} is the temperature of the surrounding air, w is the relative velocity of the aircraft and C_p is the specific heat capacity of the air.

If the efficiency for power generation on board the aircraft (or "APU"), is estimated to 25%, the combined refrigeration compressor and electric motor efficiency is estimated to 76%, then the $\eta_{Fuel\text{-}cool}$ for the vapor compression cycle is shown in fig 3. The independent variable is the condensing temperature T_1. The assumed subcooling is 5°C, superheat is 5°C and evaporation temperature is 10°C. The Joule cycle is shown below just for comparison.

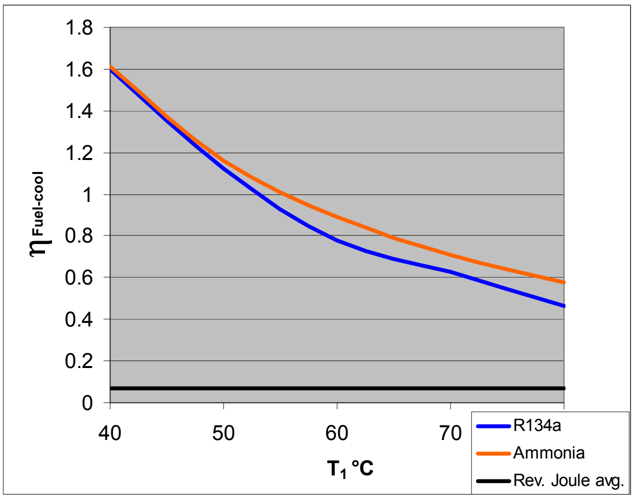

Fig. 3 $\eta_{Fuel\text{-}Cool}$ for the vapor compression cycle with evaporation at 10°C.

Without going into detail it can be seen from figure 3 that $\eta_{Fuel\text{-}cool}$ for a vapor compression cycle is a factor 10 to 20 times more efficient than the Joule cycle. The concern is the relatively heavier components of vapor compression cycle systems.

A low budget project at KTH has investigated the possibilities to make an extremely light direct cooling machine for aircrafts [4]. This will be described under 2. below.

The indirect vapor compression system, uses a secondary refrigerant. It will most likely be heavier and less efficient – but easier to fit into the aircraft. An indirect system could also easier use alternative places for dumping the condenser heat – for instance in the fuel.

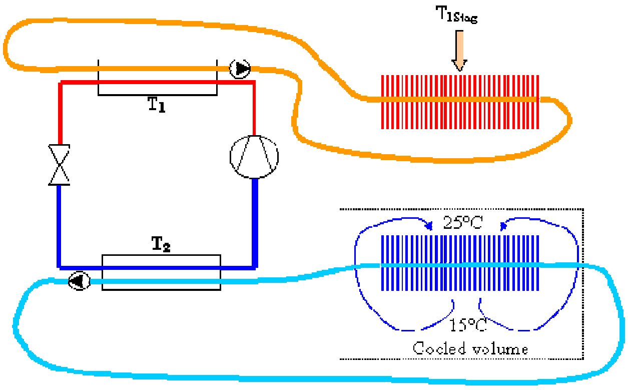

Fig 4. An indirect system with two secondary loops

2. AVAILABLE COMPONENTS

This project is primarily asking how a light cooling machine can **be built with a reasonable cost.** This often means using material from the mobile market. There will be a discussion under point 4 about visions for the future, not so limited by cost.

2.1 Available compressors

In this study the goal was to achieve 10 kW of cooling power. This can be achieved with a number of different compressors, using alternative refrigerants and running the compressors at various speeds and evaporation temperatures. This study can only analyze some possible candidates.

Some compressors are integrated with the motor into a so called hermetic unit. The advantage of such a construction is that the leakage of refrigerant is minimized. The disadvantage is that the weight for the encasing is slightly larger.

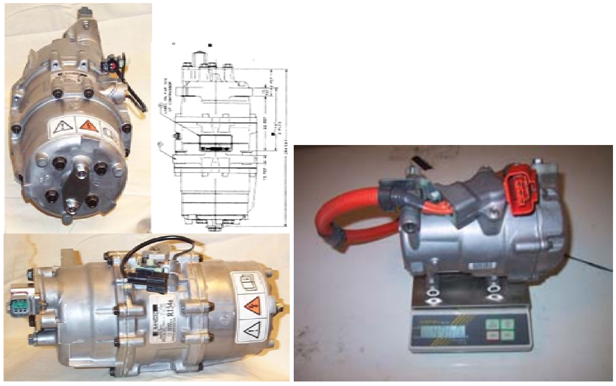

Fig. 5 A Sanden hermetic scroll motor/ compressor weighing 9.65 kg and a smaller Denso-Prius motor/compressor, to the right, weighing 4.76 kg.

Both the compressors above require an additional electric power drive which has a weight that is not negligible. The Sanden compressor has been tested in the KTH laboratory.

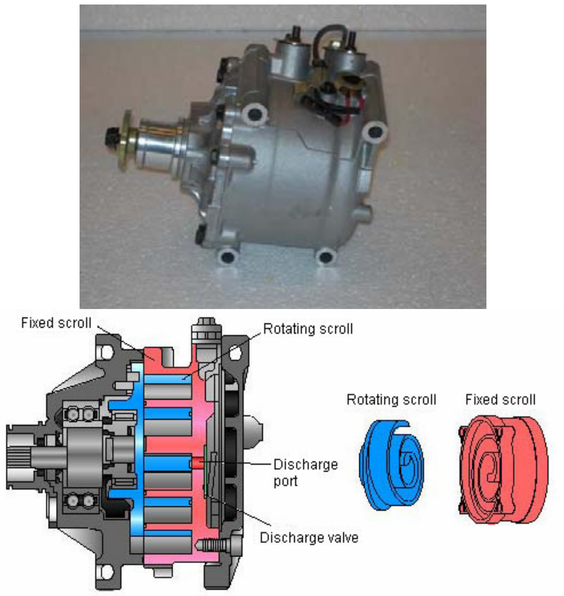

Fig. 6 An open Sanden scroll weighing 3.5 kg, 90 cc compressor above and a Visteon scroll weighing 6.5 kg **with clutch** below.

In figure 6 above two open scroll compressors for the automotive market are shown. For the experiment [4] the Sanden TRS90 compressor above was choosen. In fact both compressors are a bit too large for our purpose while running with R134a - therefore lighter open compressors like a new Sanden 50 cc will be investigated.

There are also many other methods of compressing gas like the piston, swash plate principle. However the scroll principle seems to result in the totally lightest constructions today.

2.2 Available motors

If an open construction is chosen – a separate motor is needed. There are many parameters determining the motor choice like voltage, frequency, brushes/brushless e t c. The requirement is also that the motor must fit the chosen compressor with respect to torque and speed. After the choice of a small Scroll Compressor was made, a speed of 7-10 000 rpm seems reasonable for the motor. If the cooling capacity requiremen is 10 kW a 5 kW motor would be appropriate.

The voltages 270V or 2 x 270 V DC, have been discussed for use on board future aircrafts. However no motor-controller or motor with a reasonable price have been found for this voltage. Examples of motor types considered are shown below.

Fig 7a LMC brushed motor 3 kW, 3 kg, 6000 rpm

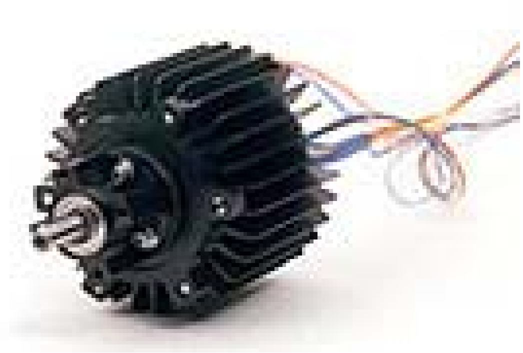

Fig 7b Aveox PM-motor 6 kW, 3 kg at 10 000 rpm

Fig 7c (Chosen) Torcman Monster 5.5 kW, 1.5 kg at 7000 rpm (www.torcman.de).

Fig 7a shows a brushed motor. Such motors are no longer preferred partly due to the risk to sparks and partly that the maintenance is costly. Fig 7b shows a, from many aspects, very interesting motor adapted for the air industry but sadly at a high price.
Fig 7c shows the chosen motor for the experiment [4]. It is normally used in model gliders and is made as an outer rotor motor. This means that the permanent magnets are located inside the rotating part and that the windings are static in the centre. The number of poles for this motor is 14. The motor in general is controlled by one light weight controller. In our case however this was a prototype motor and it had two separate sets of windings and two controllers which later proved hard to handle.

For the hermetic compressors in figure 5A controllers came with the compressors. Sadly they were either heavy or difficult to handle due to missing documentation.

2.3 Heat exchangers for handling air

Heat exchangers for airborne applications must be extremely light. A very promising technique for light heat exchangers is the extruded aluminum micro channel technique. The hydraulic diameter of the channels has today decreased to typically a millimeter, enhancing the heat transfer inside the tubes.

Another requirement for air borne applications is that the cross section area for the air flow is small. This is of course necessary for keeping the volume of air ducts as small as possible. Different types of condensers, evaporators and liquid air coolers and liquid air heaters are shown in fig 8a – 8c.

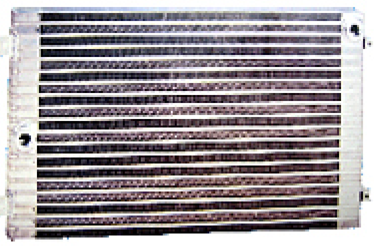

Fig. 8 a Extruded aluminum heat exchanger for the automotive industry from Showa. (http://www.sdk.co.jp/html/english/group/sdk/division/aluminium/hex.html)

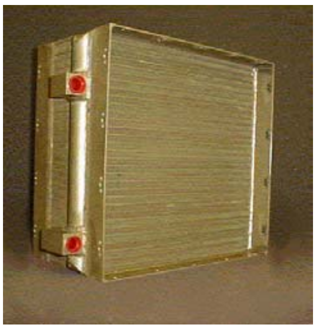

Fig. 8b Aluminum condenser for a helicopter. (http://www.tempinc.com/condense.htm)

Excellent offers for custom made heat exchangers was achieved from for example [3].

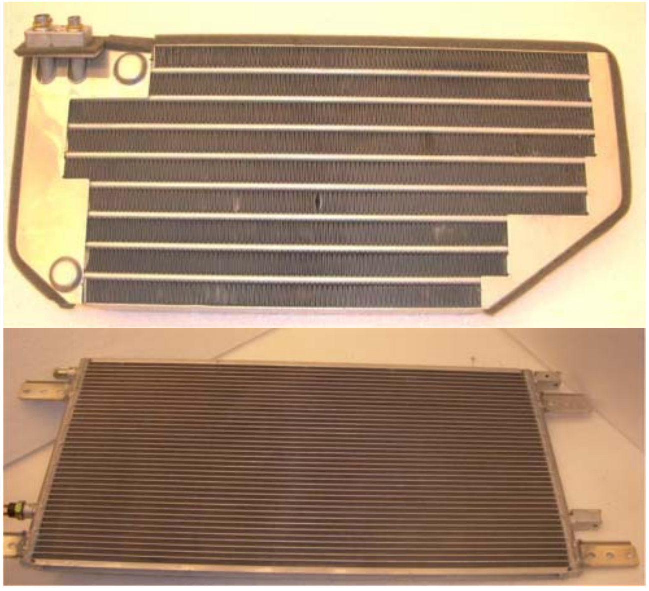

Fig. 8c (Choosen) Evaporator above and condenser below from Scania.

In the end we were kindly supplied with an evaporator and a condenser from Scania used intrucks. The nominal powers for the evaporator and condenser was 10 kW and 15 kW respectively. The weight was about 2,75 kg each. The shapes were not ideal for airborne applications but could be accepted for a test rig.

2.4 Condensers and evaporators for liquid

In an indirect system a secondary fluid (brine or coolant) is heated or cooled and then transported to the final heat sink or source. There are several reasons for using an indirect cooling system instead of a direct – in spite of that the weight increases. The AV-8 radar aboard the F-18 is for instance cooled by "Coolanol". For various reasons sometimes that fluid is being replaced by Polyalpha Olefin (PAO). In the SAAB JAS39 Gripen, PAO is used.

In an airborne application heat could for instance be given off to the fuel using a secondary refrigerant. When cooling electronics a secondary refrigerant could selectively cool those devices requiring a high cooling power. Radar equipment is one example. The cooling machine itself can also be placed more freely inside the fuselage when an indirect system is used.

Also for liquid heat exchangers the micro channel technique seems promising. Figure 8 shows the buildup of a 5 kW evaporator/ condenser before assembly at the Royal Institute of Technology.

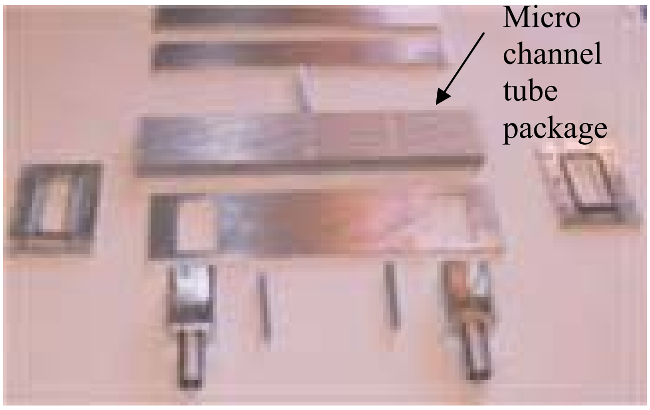

Fig. 8d Micro channel condenser/evaporator for liquid

Figure 9 shows three "generations" of evaporators/condensers developed for low weight and low refrigerant content during experiments at the lab at the Royal Institute of Technology. These heat exchangers were made for a somewhat lower power than 10 kW of cooling and 15 kW of heating. However when considering the cost for weight in airborne applications, maybe the higher temperature difference that would arise when using this size could be accepted.

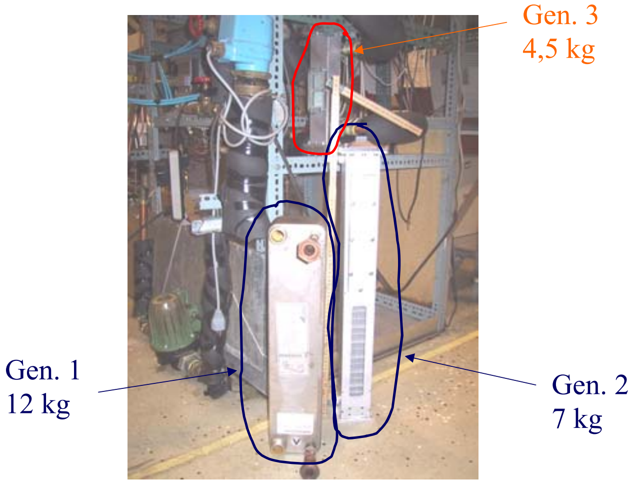

Fig 9. Three "generations" of condensers/evaporators.

The first generation is a stainless steel plate heat exchanger [1] and the 2:nd and third generation are developments of micro channel heat exchangers.

2.5 Expansion valves, fans and other devices

The expansion valve would typically be electronically controlled. There are many such valves available. One from Danfoss is shown in figure 9 below. The weight of such a valve is 0.38 kg without coil.

Figure 10 – An electronically controlled PWM modulated expansion valve from Danfoss AKV-10. (http://www.danfoss.com/Products)

For a number of reasons (economy, meassurmet stability e t c) another valve was chosen for the rig experiment [4].

Fans and pumps should also be addressed. The heat generated on the hot side in the condenser can be ridded by ram air; the air could also be driven by ejection from the motor outlet. On ground without main engine a fan will be needed. Alternatively the heat could be dumped into the fuel. Then the fan weight would be replaced by a pump weight. The ability of the fuel to absorb the heat is however limited.

On the cold side however a fan will be needed most certainly at least for a large part of the cooling need. It is necessary that these fans can handle a relatively large pressure drop as the evaporators are presumed to be thick and have a small cross sectional surface while still running with a high airflow. Examples of fans for such purposes are shown in fig 11.

Fig. 11 High speed fans from Ametek (http://www.aircontroltechnologies.co.uk/)

If a secondary refrigerant is used it could be helpul to mount the electronics directly on or close to cooled bodies. Cooled plates are shown in fig 12.

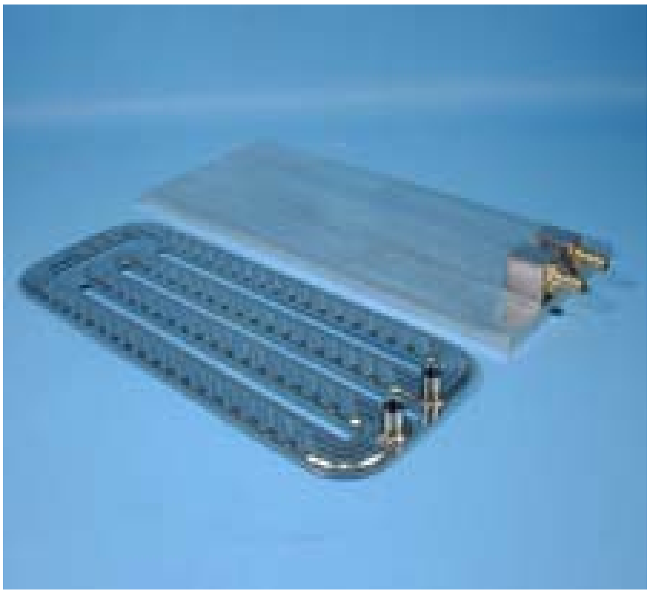

Fig 12 Cooled plates from Tykoflex (www.tykoflex.se)

3. The refrigerant

Aiming at an evaporation temperature of maximum 20°C and a condensing temperature of 80°C, refrigerant R134a can be used. A more sophisticated choice would be R124 which runs with slightly lower pressures. R124 is used in some "cooling pods". R123 would be good for cooling certain types of electronics directly by spray-cooling.

4 The test rig and future works

The basic test rig is shown in figure 13 and the assembled motor compressor is shown in figure 14.

Fig 13 the test rig during assembly

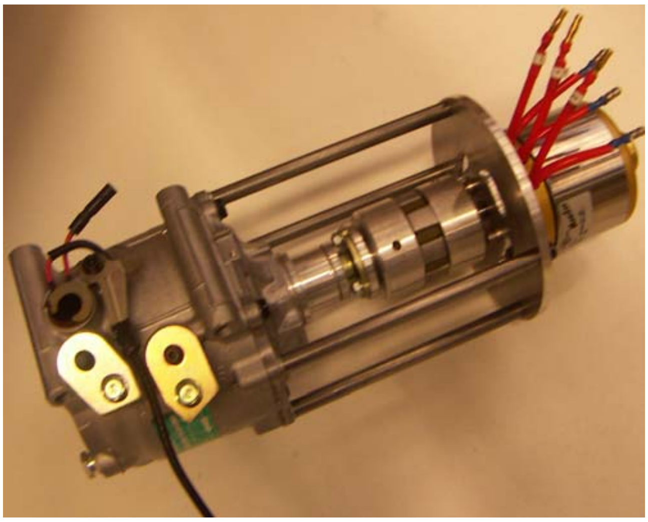

Fig. 14 An assembled motor compressor

During the tests difficulties with the motor controller was experienced. The tests will therefore be performed during 2006. Additional weight savings can be expected.

For a direct cycle it is expected to get down to about 15 kg for a cooling power of 10 kW with 20°C evaporation temperature. The components included in the weight are the compressor, motor, condenser, evaporator, expansion valve and the power electronics to run it.

References

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