GIFFORD-MCMAHON REFRIGERATOR WITH SPLIT COLD HEAD

H.-J. Forth, R. Heisig, H.-H. Klein
LEYBOLD-HERAEUS GMBH, Cologne, FRG

1. Introduction

The reason for the development of a split cold head has been the following problem:

In a vacuum vessel that consists of two separable hemispherical bowls a hydrocarbon-free vacuum of about 1.10^{-7} mbar (1.10^{-5} Pa) has to be produced. Due to the desorption rate of components situated inside the vessel an effective volume rate of flow of 10.000 ltr/s for H_2O, 500 ltr/s for N_2 and 1.000 ltr/s for H_2 is required for producing that pressure. The vacuum vessel is situated inside a magnet with a maximum magnetic flux density of c. 4 T. Access to the vessel is rather narrow and restricted. To each of the two cups a straight pipe of 5 m length and an inside diameter of c. 62 mm is fitted, but only one pipe is available for evacuation of the vessel. Because of its low flow conductance this tube cannot be used as a high vacuum line but only as a fore vacuum line. Consequently the high vacuum pump - whatever type this may be - must be situated inside the vacumm vessel itself, as a sort of integrated pump. Various types of pumps have been taken into consideration: The getter pump with water- or LN_2-cooled shield, the cryopump with LHe or refrigerator cooling. With respect to the relatively long operating periods of several weeks as required by the respective project all pumps that require LN_2 or LHe for their operation had to be ruled out. Discussion settled on a particular type of cryopump that is cooled by a two stage cold head of a Gifford-McMahon refrigerator.

2. Requirements on the cryocooler

In order to meet the requirements with respect to pressure and volume flow rates (and hence temperature) as given in section 1 the refrigerating capacity of the cryopump must be 10 W at 80 K (first stage) and 2 W at 20 K (second stage), and the ultimate temperature must be less than 14 K. The distance between compressor and expander (refrigerating part of the cold head) must be at least 5,5 metre,the strong magnetic field of 4 T must be taken into

consideration and no other electric and/or magnetic fields should be present. Other requirements are: small size, high reliability, long operation periods, low price etc. These requirements are similar to type of cryocoolers as used for cooling electronic devices.

Application to our special case implies:

- use of a split-cycle cryocooler
- small dimensions are required only for the expander assembly
- energy consumption is not particulary interesting
- low overall costs are important (investment, operating costs and maintenance)
- reliable operation with no maintenance for 8000 hours (1 year).

3. Checking the suitability of known devices

The following known devices which have separate expanders were investigated: the split Stirling cooler, the split Vuilleumier cryocooler and the Gifford-McMahon (GM) cryocooler.

a) The split Stirling cryocooler has a great advantage:
it has no valves and only a single connecting line between compressor and expander. But this type of cooler reacts in his efficiency rather sensitive with respect to the length of the connecting line and the void volume. After Chellis (1) 5 feet long lines are possible, but lengths of 12 to 18 inches are practicable.

b) the split Vuilleumier cooler is virtually a Stirling cryocooler wherein the mechanical compressor component has been replaced by a thermal compressor. Thus the length of the connecting line is limited in analogy to that of the split Stirling cryocooler.

c) The Gifford-McMahon cryocooler has a widely separated compressor assembly and expander assembly (also called compressor unit and cold head). Two connecting lines join them. The cold head comprises a small motor to operate the valves and in some types the same motor moves also the displacer; other types have a gas drive.

The compressor unit may be seen as a gas source with nearly constant pressure levels. The valves isolate compressor unit and cold head from one another and allow that the connecting lines can be of almost any length. Their volumes are part of the gas reservoir of the compressor unit.

In conclusion: The split Stirling and split Vuilleumier cryocoolers had to be dischardedbecause of the long distance required between compressor and expander, the GM-cryocooler in its normal version because of the motor for the valve control. However, modification of the GM-cryocooler appeared to be feasible.

4. Ways of solving the problem

In view of the required operation of the cooling device within the strong magnetic field the   usual valve control by means of a small electric motor had to be abandoned or moved outside the inaccessible zone or integrated in the compressor unit. The displacer movement has to be effected by means of a pneumatic drive.

As any type self-regulating operation is not available - in spite of several well known patent applications - and as - generally speaking - such self-regulation appears to be rather sensitive with respect to gas supply, gas consumption and friction of the sealing elements, we have deceided to displace valve control device and to split the cold head into expander and valve control device.

4.1 Solution

Fig. 1 shows diagrammatically various arrangements of the GM-cryocooler

Fig. 1a   - normal GM-cryocooler

Fig. 1b   - GM cryocooler with splitted cold head, with the valve
            control part outside the inaccessible zone

Fig. 1c   - as Fig. 1b, but with the valve control part integrated
            in the compressor unit.

Based on the required refrigerating capacity - see section 2 - the standard LH Cryorefrigerator R 210 was used; this refrigerator has the cold head RG 210 with refrigerating capacity of 12 W/80 K and 2 W/20 K for the first and second stage respectively; the compressor unit type RW 2, 60 Hz has an electric power consumption of 1,5 kW. The maximal diameter of the first stage is 55 mm, which allows the expander to be built inside the fore-vacuum line with an inner diameter of 62 mm. The unit therefore becomes 30 % shorter and has by more than 30 % less weight as compared with the standard cold head.

4.2 Testing the realization

Is the solution described above in a position to satisfy the requirements mentioned at the beginning of this paper? To what extent do the mainparameters: length of connecting lines and void volume influence the behaviour of the device? Critical parameters are: Filling and draining of the expander, the effective pressure difference Δp_{eff} in relation to the pressure difference Δp_{cu} of the compressor unit. Furthermore:

- the heat of compression during filling of the expander and
- heat abduction from the expander.

4.2.1 Effective pressure difference Δp_{eff} in the expander

The filling and draining procedure of the gas filled volumes of the expander of GM-cryocooler with splitted cold head can be described as a filling of the expander volume from endless volume with constant pressure on high level through connecting line as a flow resistor and as a draining of the expander volume into the another endless volume with constant pressure on low level etc. The changes from low to high pressure happen very fast.

Already the qualitative comparison of the filling of the GM-cryocooler with splitted cold head with the filling of the expander of the split Stirling cryocooler occurs than in the case of the split Stirling cryocooler, more complete because of the virtually sudden change of the pressure in the case of split GM cryocooler.

The amount of gas required by the cold head RG 210 (see above) is about 20 stdm³/h. The additional consumption of a line of for example ISO-size 4 amounts theoretically to 6.3 stdm³/h. The higher gas requirement or - in other words - the smaller pressure difference $\Delta p = p_{high} - p_{low}$ of the system is partly compensated by the improved efficiency of the compressor. The applied measures might compensate the losses and increase the relative refrigerating capacity from 76 % to 84 % for this case.

4.2.2 Heat abduction from the expander

The nominal refrigerating capacity of the used cold head RG 210 amounts to 12 W/80 K plus 2 W/20 K; if one adds the unavoidable losses in the regenerator then the total thermal load on warm end of the expander amounts to c. 20 W. In the case of a normal GM-cryocooler this heat is removed by the discharged gas. A cold head RG 210 has a temperature difference between high pressure and low pressure gas of about 4 K.

Things are much different in the case of cryocooler with split cycle and of GM cryocoolers with splitted cold head. The thermal load causes a warming up of the gas pipe and the expander housing.

Two ways are available for stabilising the system:

- additional air- or watercooling of the warm end of the expander
- introduction of a forced gas circulation as in normal GM cryocoolers by installation of a separate gas filling line connecting to high pressure valve and gas draining line connecting to low pressure valve (This however involves doubling of the void volume of the lines and corresponding losses as discussed above).

5. Experimental set-up

The simple set-up is shown in Fig. 2. The working and the control space of the cold head are supplied with gas via separate pipes. For the given length of line of 5,5 meter the following parameters were varied:

- inner diameter of the line leading to the working space: ISO-size 8, 6, 4, 2 mm
- inner diameter of the line leading to the gas drive: ISO-size 6, 4, 2, 1 mm
- cooling of the warm end of the expander
  o air cooling with free convection
  o air cooling with forced convection
- watercooling with forced circulation
- gas filling and gas draining through separate lines, connecting to high and low pressure valves

6. Experimental results

6.1 Dimension of the connecting lines

The experiments showed that the rating of the connecting line to the working space is determined by the small void volume requirement.

The void volume was smaller than the geometrie volume of the expander, the diameter was ISO size 6. For the diameter of the connecting line to the gas drive a diameter of ISO size 2 was found which ensures the pressure increase and fall and the correct timing of displacer motion.

6.2 Heat abduction from the expander

If the thermal load causes the temperature increase of the warm end of the expander, the refrigerating capacity of the first stage will be drastically reduces.

The influence of temperature of the warm end of the expander on the refrigerating capacity of the first stage at 80 K and second stage at 20 K give the Fig. 3. Cooling is necessary.

6.2.1 Air cooling

Cooling by air with free convection was not sufficient, the max. temperature of the warm end was nearly 85 °C. The refrigerating capacity was too low.

Air cooling with forced convection by fan was able to drop this temperatur level below 40 °C. The refrigerating capacity reached nearly 50 % and 80 % respectively of the nominal refrigerating capacity of the first and second stage. One bigger heat exchanger could not be realised (s. section 2).

6.2.2 Watercooling

Watercooling with forced circulation of the warm end of the expander showed the best results. The results obtained when watercooling, the expander housing or the end of the connecting line to the working space or both parts were only slightly different. The first and second stage refrigerating capacity of this GM cryocooler with splitted cold head reached 80 % and 85 % respectively of the nominal refrigerating capacity of the same GM cryocooler working in standard configuration.

6.2.3 Gas filling and draining through separarte lines

The losses caused by the void volume which is twice that of prevoius design, go-up in proportion to the void volume; the losses caused by the self-warming-up of the end of the expander can be neglected. The improvement of the efficiency of the compressor increases the relative refrigerating capacity of the first and second stage to 70 % and 82 % respectively of the nominal refrigerating capacity of the same GM-cryocooler in standard configuration.

The refrigerating capacity of the first and second stage of splitted cold head RG 210 is shown for the discusses arrangements at Fig. 4.

7.   Conclusion

Leybold-Heraeus Co. have developed, built and succesfull tested a Gifford-McMahon cryocooler with splitted cold head for cooling a cryopump.

The refrigerating part of the cold head and the gas flow control device have been separated (splitted cold head) and the distance between them is bridged by only two thin lines for carrying the working gas. Due to this separation the size of the refrigerating part is virtually defined only by the size of the displacers whilst the gas flow control device can be of any desired design.

It has been shown that dimensioning of the connecting lines and the corresponding losses became less critical with increasing size of the expander, but additional cooling in proportion to the refrigerating capacity is required.

Such small and light-weight cryocooler, which does not produce any magnetic and electric fields nor is affected by strong magnetic fields appears ideal to be used:
as single-stage cryocooler for sensors to approx. 30 K (e.g. infrared detectors),
as two-stage cryocooler for objects down to approx. 10 K (e.g. cryo-pumps) and
to precool Joule-Thomson-cycle achieving temperature down to 4,2 K (e.g. SQUIDs, Josephson-junction-elements).

8. References

1) F. F. Chellis: Design Compromises in the Selection of
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2) H.-J. Forth, R. Frank, R. Heisig, H.-H. Klein: A new Develop-
   ment of Refrigerators of high operational
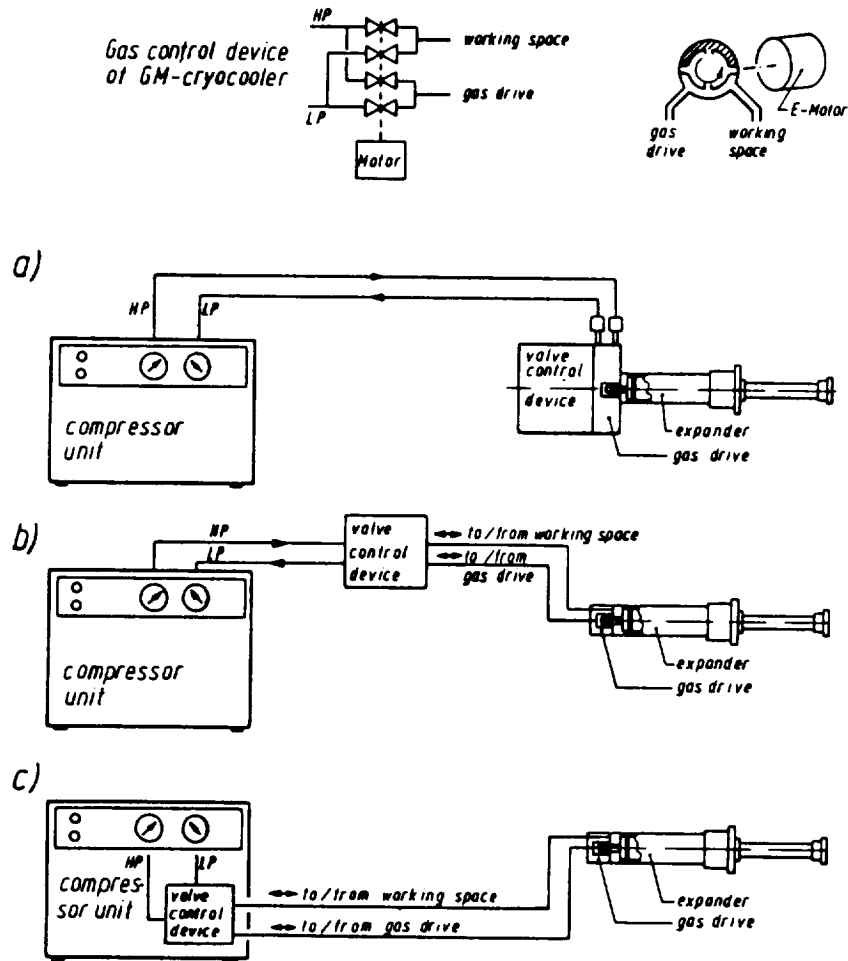   Reliability for Use in Cryopumps
   IVC, Cannes 1980

Fig. 1: Diagrammatically various arrangements of the GM-cryocooler.

Fig. 1a - normal GM-cryocooler.

Fig. 1b - GM cryocooler with splitted cold head, with the valve control part outside the inaccessible zone.

Fig. 1c - as Fig. 1b, but with the valve control part integrated in the compressor unit.
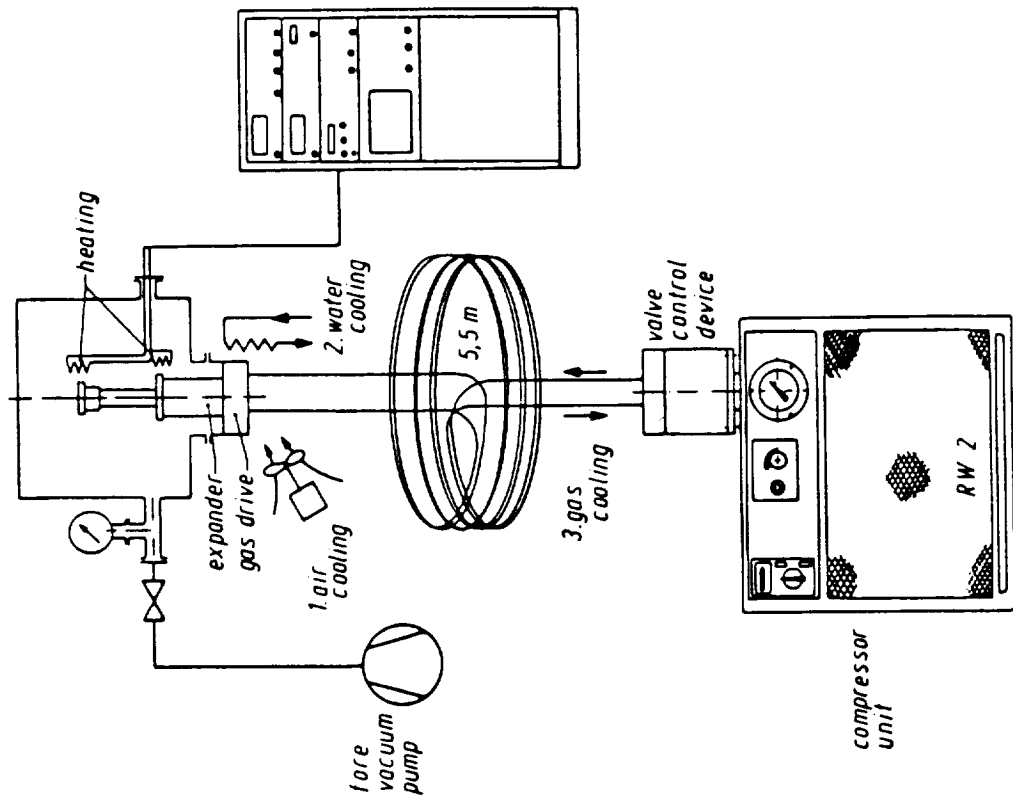
Fig. 3 graph labels:

water cooling

air cooling with fan

air cooling

$\dot{Q}_1$ first stage / T_2 = 80 K

$\dot{Q}_2$ second stage / T_2 = 20 K

$\dot{Q}$

10 W 8 6 4 2

300 320 340 360 K — 20 40 60 80 °C

T_E

whereby $\dot{Q}$ refrigerating capacity of splitted cold head
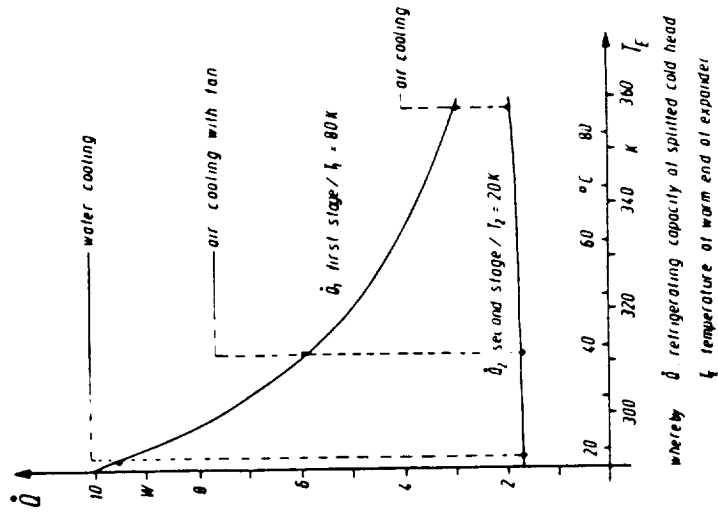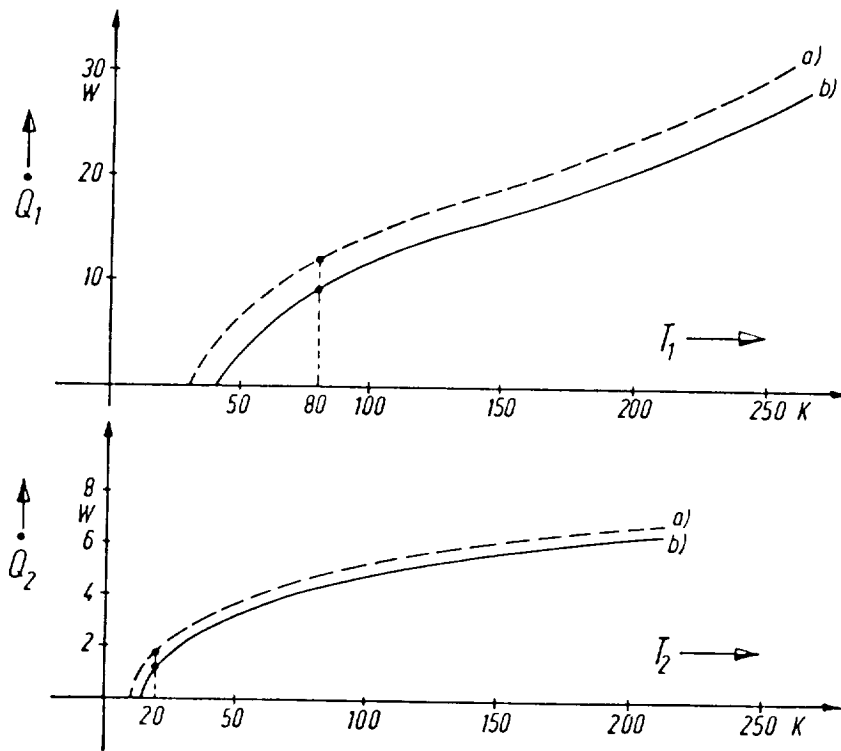
T_E temperature at warm end of expander

Fig. 3: Refrigerating capacity $\dot{Q}_1$ (80 K) of the first stage and $\dot{Q}_2$ (20 K) of the second stage of GM-cryocooler with split cold head independent of temperature T_E of warm end of the expander

Fig. 2 labels:

heating

2.water cooling

5.5 m

expander gas drive

1.air cooling

3.gas cooling

valve control device

compressor unit

RW 2

fore vacuum pump

Fig. 2: Experimental set-up.

RG 210   a) standard cold head
          b) splitted cold head $T_E \approx 300K$

whereby: $\dot{Q}$: refrigerating capacity
          $T_{1,2}$: temperature of the stage
          T_E : temperature of warm end of expander

Fig. 4:   Refrigerating capacity $\dot{Q}_1$ of the first stage and $\dot{Q}_2$ of the second
          stage in dependance of their temperatures for standard cold head
          and splitted cold head RG 210