PERIODICA POLYTECHNICA SER. TRANSPORTATION. ENG. VOL. 20, NO. 1, PP. 73-80 (1992)

COLD AIR REFRIGERATING MACHINES WITH MECHANICAL, THERMAL AND MATERIAL REGENERATION

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Presented by Prof. E. Pásztor

Received: April 5, 1990.

Abstract

The cold air refrigerating machines are practically of the same age as the vaporisation-compression-type refrigerating machines, however, the former ones were put in actual operation only during the past decades. As the principal reason for it, their lower economical efficiency can be considered. However, the vaporisation-compression-type refrigerating machines cause heavy pollution to the environment (gap of ozone layer problem), therefore the interest in the cold air refrigerating machines has been greatly aroused lately. In the paper, the operation, theory and application possibilities of the cold air refrigerating machines are dealt with, especially considering the air- conditioning of motor vehicles.

The improvement of the economy of cold air refrigerating machines is especially paid great attention in this paper. In addition, the possibilities of applying heat exchangers, as well as the cooling process of the operating-medium during compression with the help of liquid charge are examined.

Keywords: theory of cold air refrigerating machines, application of heat exhangers, cooling during compression.

Introduction

Today the mechanical generation of the energy form 'cold' required for food conservation in stationary and mobile systems and for refrigeration technology purposes is effected almost exclusively by compression-type refrigerating machines (KDKM). Their high technological standard is the result of an intensive engineering development which was initiated in 1874 by the ammonia compression refrigerating machine invented by LINDE. Within a very short time it replaced almost completely the cold air refrigerating machine (KLKM) which dated back to HERSCHEL (1834), was constructed for the first time by GORRI (1844) and developed ready for industrial production by KIRK, WINDHAUSEN and GIFFARD. The reasons for this trend were the latter's energetic inferiority and its considerably greater floor space occupied at that time.

Owing to the restrictions on chlorinated fluorocarbons (CFC) to protect the ozone layer having come into force since January 1, 1989 after the ratification of the Montreal protocol of UNEP, refrigeration technology branches suddenly find themselves confronted with a great number of difficult problems which have to be solved within a short time. A main problem is that the thermodynamic parameters of the known 'CFC' substitutes with their lower ozone hazard potential are worse than those of the safety refrigerants now being used in compression-type refrigerating machines which indeed have a very high ozone hazard potential ($R11$, $R12$ ...). The possible alternative refrigerants which are ecologically more beneficial require a number of technical changes of the conventional KDKM's to compensate for these disadvantages, Therefore, extensive scientific and engineering work is necessary and parallel thorough studies of cold-air refrigerating machines are now justified. There is much to be said in favour of these studies, especially the fact that the working means of KLKM's, being cold air, are not at all dangerous to the ozone layer and that today more technically mature machinery, equipment and instrument systems are available. Due to these favourable prerequisites the chances of success of KLKM's for higher temperature use, as in air-conditioning, chilled and frozen goods storage, ought to be now greater than in the past. These chances are favoured by a two-stage KLKM according to [1] with mechanical, thermal and material regeneration, if the thermodynamic process which is outstanding for its considerably improved energy balance can be satisfactorily converted with a view to its technological aspects.

Description and Evaluation of the Cold Air Refrigerating Machine

Circuit Diagram and Cycle Processes

The circuit of a KLKM shown in *Fig. 1* according to [1] would permit a noticeable improvement of the energy balance, even when used in a higher temperature range. It is a two-stage system with mechanical, thermal and material regeneration. Mechanical regeneration is realized in the first compressor stage. In the supercharger 2 driven by the expansion turbine 9 through mechanical coupling a recirculation or mixed air flow can be drawn in at the suction place 1 under an ambient pressure of $p_{a}=p_{\min}$. This flow can then be precompressed to a clearance pressure p_{zw} and subsequently cooled back in the first cooler 3 to an ambient temperature T_a. It has proved favourable to use radial machines for the turbine and the compressor mounted on a common shaft. For the second compression stage a waterflooded screw compressor is used to which the KLKM drive energy can easily be fed from the outside via a drive system 5. Air compression in it is made by water injection at almost the same temperature. The necessary water is extracted from the wet air compressed with maximum pressure in a second cooler 6 arranged after the screw compressor, the water falling below the dew point temperature and being collected in a pressurized water tank 7. From here a portion of the water is fed to the screw compressor via a control number 8 and another portion to the mixing chamber 10 of the KLKM by taking advantage of the existing pressure gradient. This process designated as material regeneration in the second stage, thus reducing considerably the drive energy and increasing the performance coefficient of the KLKM. On the basis of material regeneration it will be possible at the same time to operate turbine 9 by means of dried air and thus to avoid operating troubles due to icing when the air is expanded. Furthermore the dry cold air from the turbine can again be wetted in the mixing chamber with the water at first extracted from the air and then it can be prepared together with the recirculation air from store 12 to form the necessary air supply flow. By means of thermal regeneration in a recuperator 13 arranged after the cooler 6 the performance coefficient of this KLKM is further improved. The thermodynamic cycle process for this circuit (in *Fig. 2*) shows that compared with the two-stage anti-clockwise Joule process with adiabatic irreversible compression the performance coefficient is increased among other things by the nearly isothermal compression in the second stage.

Computation Model and Evaluation

The energetic evaluation of the KLKM described in *Fig. 1* is made by means of performance charts [2]. As regards their content they are a graphical representation of the functional relation between driving and refrigeration performances for any parameter combinations.

The function of the driving performance of a KLKM is shown in the general *Eq.* (1).

$$\begin{aligned} P_A = f \Big\{ & \dot{Q}_0,\ \dot{m},\ \pi,\ \eta_{VV},\ \kappa,\ n,\ \vartheta_3(T_a,\ \Delta T,\ T_i), \\ & \Delta\vartheta_{\mathrm{REG}}(\Delta T_{\mathrm{REG}},\ T_i),\ \eta_{iT}\left[\dot{V}_3,\ (\dot{m},\ \pi,\ T_{3\prime})\right], \\ & \eta_{iVV}\left[\dot{V}_1(\dot{m},\ T_1)\right],\ \eta_{iNV}\left[\dot{V}_{1\prime}(\dot{m},\ \pi,\ T_{1\prime})\right] \Big\}\,. \end{aligned} \tag{1}$$

The computation is based on the prerequisite that the energy demand of the supercharger is met by the expansion turbine.

Eq. (1) shows the dependence of the driving performance P_A upon the refrigeration performance Q_0 and the process parameters, as partial

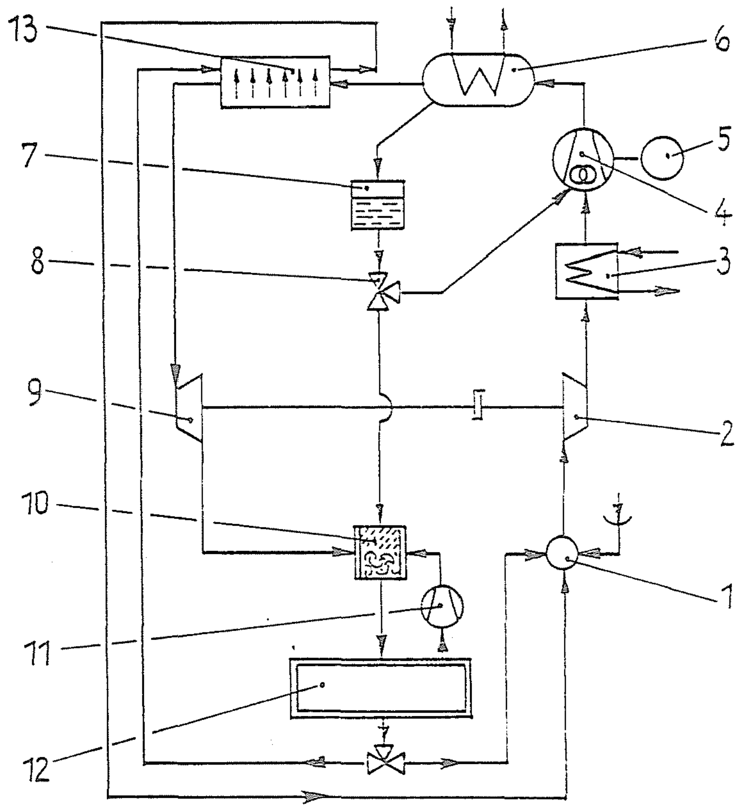

Fig. 1. Diagramm of a KLKM with mechanical, thermal and material regeneration

Terms:

1 suction place	8 control member
2 supercha	9 expansion turbine
3 radiator	10 mixing chamber
4 re-compressor	11 fan for recirkulation air
5 drive system	12 cold store
6 consender	13 recuperator
7pressurized water tank	

air flow $\dot{m}$ [3], total pressure ratio π as well as outside and inside air temperatures T_a and T_i. Included are the thermal regeneration degree $\Delta\vartheta_{\mathrm{REG}}$ (or ΔT_{REG}), the mechanical regeneration through the pressure ratio in the first compression stage π_{VV}, the material regeneration through the polytropic relation $1 \leq n \leq \kappa$ and also the volume flow-dependent isentropic

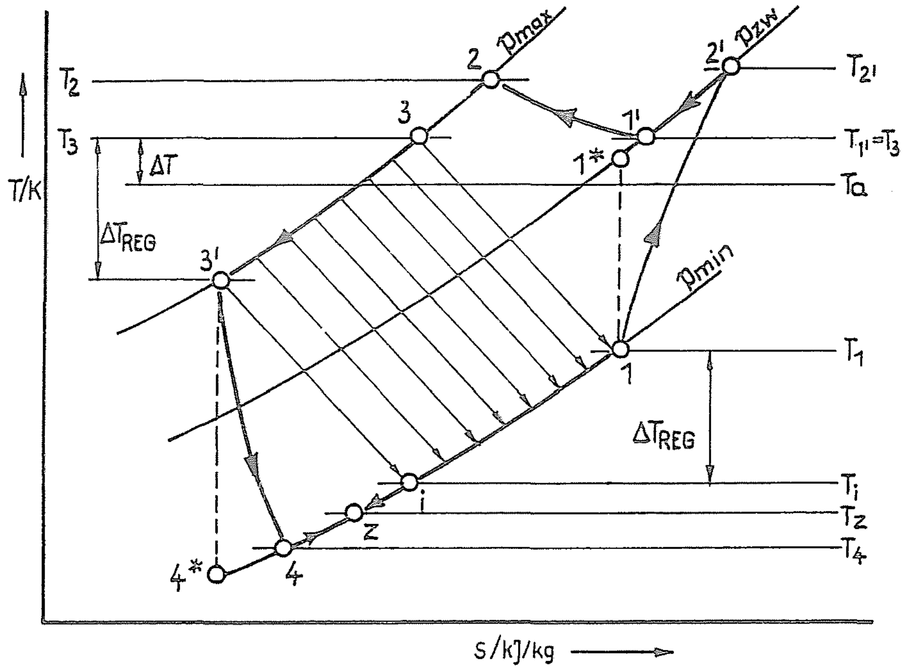

Fig. 2. Process behaviour of a KLKM working accordance to fig. 1

mechanical efficiency of the turbine η_{iT}, the supercharger η_{iVV} and the postcompressor η_{iNV} by means of empirically determined functions [2].

In the performance chart, shown in *Fig. 3,* the lines of constant performance coefficients $\varepsilon = \dot{Q}_0/P_A$ are entered. They permit not only a simple determination of energetically favourable process parameter combinations for the KLKM described in *Fig. 1,* but also an energetic comparison with existing KLKM's and conventional KDKM's having different refrigeration performances $\dot{Q}_0$.

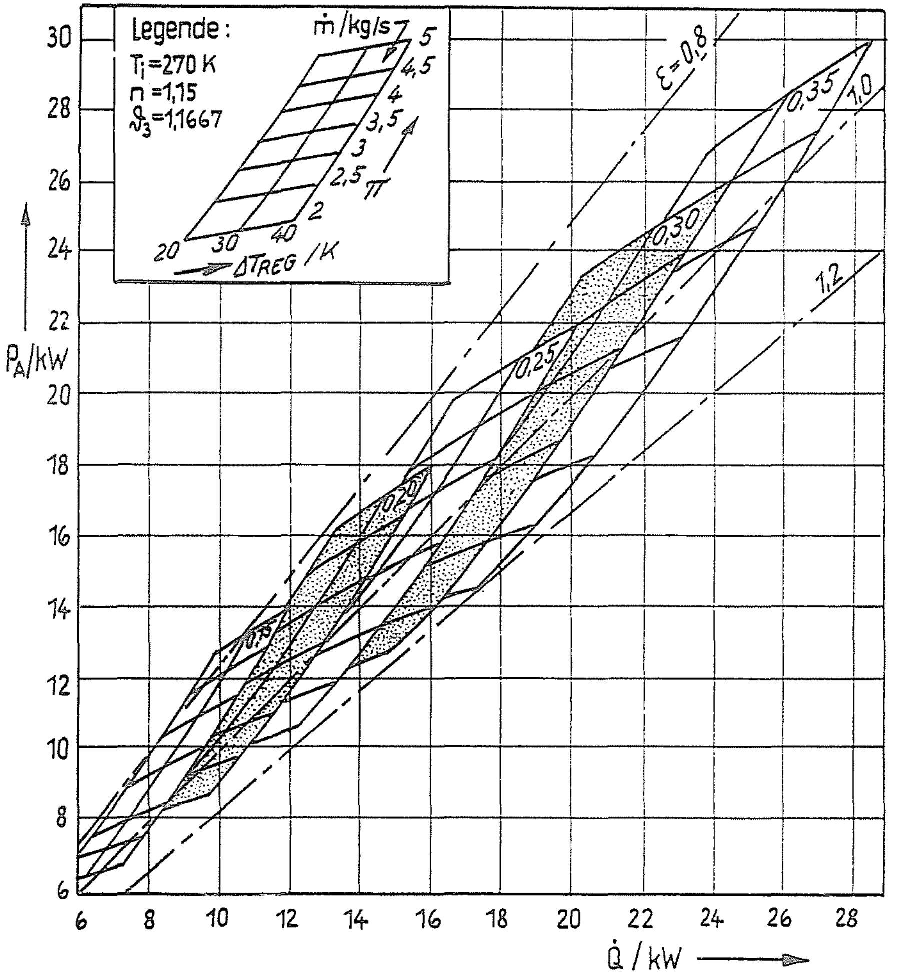

Fig. 3. Performance chart of a KLKM working according to fig. 1

The performance chart according to *Fig. 3* holds true for an inside air temperature of $T_i = 270\,\mathrm{K}$, thus including the temperature range of frozen goods storage. The polytropic exponent is held constant at $n = 1.15$, because between $n = 1.10$ and $n = 1.25$ a driving performance change amounting to $P_A = 0.5\,\mathrm{kW}$ has proved to be unimportant. This result implies that it is not necessary to make great demands on the control of water injection into the re-compressor.

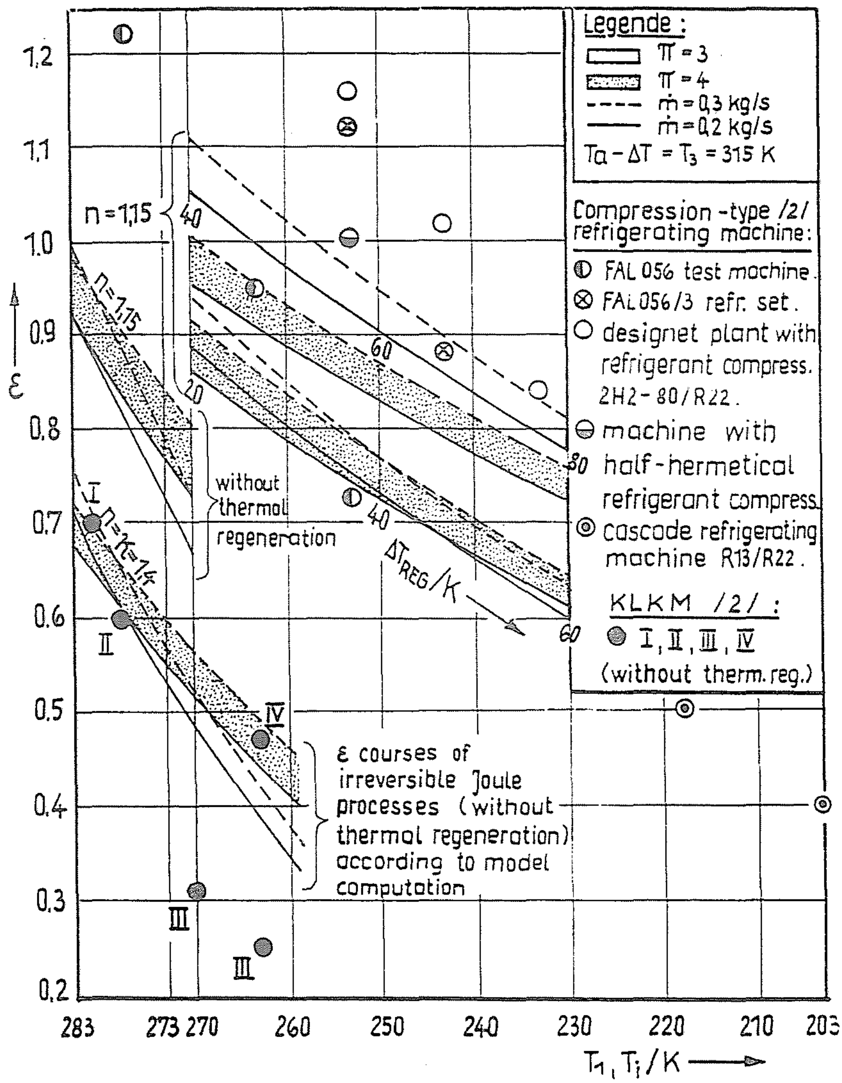

Fig. 4. Energetic comparison between existing compression-type and cold-air refrigerating machines and a KLKM working according to fig. 1

Taking the value of

$$\vartheta_3 = \frac{T_3}{T_i} = \frac{T_a \Delta + T}{T_i} = 1.1667 \qquad \text{with} \qquad \Delta T = T_3 - T_a \qquad (\textit{Fig. 2})$$

an ambient temperature of $T_a = 310\,\mathrm{K}$ is fixed.

While conventional KLKM's give performance coefficients in the abovementioned temperature range of $\varepsilon = (0.5$ to $0.7)$ at the most, the performance chart in *Fig. 3* for a KLKM working according to *Fig. 1* already shows at least values of $\varepsilon = (0.8$ to $1.2)$. The higher values of ε are obtained in the case of greater air flows $\dot{m}$ and regeneration degrees ΔT_{REG} and in the case of lower pressure ratios π. Furthermore the performance chart shows that in the interest of high performance coefficients a definite refrigeration performance $\dot{Q}_0$ is always connected with a definite air flow $\dot{m}$.

The increase of the performance coefficient together with the air flow $\dot{m}$ results from increasing isentropic mechanical efficiencies η_{iT}, η_{iVV} and η_{iNV} and increasing volume flows $\dot{V}_3$, $\dot{V}_1$ or $\dot{V}_{1'}$ entering the machines (*Fig. 2*).

On mechanical grounds it is not possible to decrease the pressure ratio below $\pi = 3.5$. Extensive studies in [2] have shown that the optimum pressure ratio is in the order of $\pi = (3.7$ to $4.3)$ being practically independent of the temperature T_i. This fact points to a rather tangible advantage of KLKM's compared with KDKM's the latter requiring a considerably higher pressure ratio and above all a far greater maximum process pressure $p_{\max}$ for equal temperature conditions T_a/T_i.

An energetic comparison between conventional KDKM's and KLKM's with mechanical, thermal and material regeneration (*Fig. 4*) shows that especially due to the high technological development state of compressors and turbines there are real possibilities to reduce to a large extent the still existing energetic drawbacks of KLKM's in the higher temperature ranges. With a view to those refrigerants presenting a high ozone hazard potential which will shortly be due to be eliminated, KLKM's ought to be a true alternative to KDKM's.

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