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Experimental Investigations on A Pulse Tube Water Cooler Using Natural Refrigerant

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Abstract

Pulse Tube Refrigerator (PTR) offers an alternative to vapour compression refrigerator designs for domestic refrigeration systems if it's COP become comparable. However applications will become a reality when efficiencies become greater than 20% of Carnot Efficiency. The requirements for a domestic refrigeration system are: simplicity, reliability, COP and low cost. With an objective of developing an alternate refrigeration system for water cooler application at 13^{o}C with a capacity of 80 W, a simulation program to quantify the performance of a Stirling type Orifice PTR is developed with nitrogen as the refrigerant and performances are predicted for the PTR. At 274 K, the refrigerating effect predicted is 12.689 W, whereas the experimental value is 11.57 W, a difference of around 9%. Experiments conducted on the PTR developed is analysed for feasibility of applications in a water cooler. A maximum refrigerating effect of 30 W is obtained for a net power input of 497 W, the COP being 0.06 at 390 rpm of compressor speed and charge pressure of 9 kg/cm^2. A Stirling type OPTR coupled to a One TR open type refrigeration compressor using Nitrogen as the working fluid operating at 4 kg/cm^2 can be developed for a 10 lts/hr water cooler. A low cost oil free open type reciprocating compressor can make the Pulse Tube Water Cooler a reality.

Keywords: Simulation, Alternate Refrigeration system, Nitrogen refrigerant, Orifice Pulse Tube Refrigerator

Introduction

Refrigerants have been the harbinger of growth and change to the world of cooling. CFC's, the standard refrigerant since 1920's was found to be responsible for ozone layer depletion. The heightened sense of urgency to achieve crucial environmental goals, has made the established patterns of refrigerant use in a state of flux. The global consumption of CFC's in refrigeration, air conditioning and heat pump sectors is 25% of the total. A leading brand of water cooler in the market with a rating of 20 lts./h, 80 W cooling capacity has a COP of 0.9. If the motor and compressor efficiencies is to be 80% and around 20% losses is due to irreversibility in a real system, the ideal refrigeration cycle using alternate refrigerants should target a cycle COP of about 2, for an early phase out of CFCs.

The efforts being expended on the development of environmentally benign systems as replacements for CFC systems, has led to competitions among various refrigerants. Along with the search for new refrigerants, research into alternative refrigeration systems will call for more attention now. Distinguishing between new and existing equipment, existing large systems will have to be upgraded and replaced slowly, taking up to 10-15 years. New design using alternative refrigerants is possible in many sub sectors. Domestic refrigerators can be one of them, because of its current dependence on CFCs, present size and rate of growth and ultimate market potential. Simplicity in fabrication and operation, minimum number of components, easy market assimilation and penetration are some of the factors needed for an alternative refrigeration system, with a non CFC refrigerant. A refrigerator must fulfil the requirements such as Capacity, Input power, Efficiency, Cool down time etc., for which it is designed. Enumeration of the losses and thermodynamic properties of the working fluid at varying pressures and temperatures will predict the actual performance of the system.

Alternatives like R134a, Hydro Carbon mixtures, propane have been considered as retrofit refrigerants in the domestic refrigerators. Propane is inflammable and R134a, a green house gas, has to be phased out by 2040 in developing countries. Helium and Nitrogen being environment friendly, with no environment consequences due to leakage of refrigerant, necessitates a look as the refrigerant alternative. Nitrogen does not change its properties due to its inert nature. During the development of a Stirling Cryocooler, Organ [1] concluded that the finding supports a serious evaluation of Nitrogen as a working fluid for CFC-free domestic refrigeration.

Pulse Tube Refrigeration, introduced by Gifford in 1961 as a method of cooling has gained interest and popularity due to its simplicity and hence high reliability. The absence of moving parts in the cold region, small compression ratios and no use of sliding seals and lubricants leads to increased reliability. In a PTR, a pressure oscillator, consisting of a compressor and a set of valves, is connected to a long tube, via a thermal regenerator and heat exchangers. Inclusion of a reservoir connected through an orifice and a double inlet method of the working fluid improves the refrigeration performance. The obstacle to its wide application is the relatively low refrigeration performance in comparison with other kinds of refrigeration. The Stirling type PTR or high frequency PTR with a minimum life of one year, has an efficiency of around 10% Carnot efficiency and costs $7000 [2]. The cost is due to the high efficiency electromagnetically driven linear compressors. In an inertance tube PTR of 6 mm inner diameter and 3.5 m length, a cooling power of 160 W at 77 K was reached, with a COP of 4.5 %. [3]. deWaele [2] deduced that the application of PTR for domestic refrigeration seems to be rather far away. Despite its shortcomings, the simplicity and consequent reliability of a PTR system gives an impetus for an investigation into its feasibility in domestic refrigeration as an alternative to CFC based systems. The advances in practical techniques of PTR development are largely responsible for the continued interest in the pulse tube.

Many methods of analyses have been employed to predict the performance of a PTR system. The enthalpy flow model, thermodynamic model, and thermoacoustic models [4] do not give a generalized optimization methodology for the design of a PTR of desired capacity. The Isothermal model proposed by Zhu and Chen [5] also does not give a realistic simulation of the Orifice type PTR (OPTR). Hence, there is a need for a realistic mathematical model, which can be used to design and optimize a PTR for a refrigerator application. A simulation program developed by the author incorporates the mathematical equations, predicts the performance of an OPTR based on given input geometric and operation parameters.

Refrigerator Simulation

The mathematical description of the processes occurring in a pulse tube and regenerator is given and the model for oscillating flow of fluid is formulated [1], similar to the method followed by Zhu and Chen [5] and Atrey [6]. It is based on the thermodynamic laws, conservation laws written in differential form and ideal gas laws. In the present model, convergence value used in the calculation of pressure is by Newton-Raphson method. Unlike Zhu's [5] model, the instantaneous pressure is taken as the sum of masses of 'cpt, hpt and mpt of the pulse tube section and HHE, leading to higher pressures in the system. Also the compression and expansion of gas in the pulse tube section is assumed to be adiabatic. The variation in properties of the working fluid and the material with temperature of the system during its cyclic compression and expansion in the system is modelled. The above assumptions and methods lead to a more realistic simulation of the working of an OPTR compared to Atrey [6] and Zhu and Chen [5] model. The equations that govern the flow and heat transfer in the regenerator are derived. The system of equations together with boundary and initial conditions, geometric and operational parameters suitable for numerical solution, is obtained. The resulting algebraic equations are incorporated in a computer program and coded for the simulation of an OPTR [1].

Adiabatic Model of PTR

The PTR works by the cyclic compression and expansion of a fixed quantity of gas, usually helium. The essential elements of a pulse-tube refrigerator are shown in Figure 1. In the present model, the volume of gas in the pulse tube and HHE is included in the calculation of instantaneous pressure for each step of crank rotation. The volume variations in the cylinder are assumed to be sinusoidal. The temperature profile of the gas in the components of OPTR is obtained from the ideal gas laws. The work done and cooling capacity is calculated based on the P-V diagram area of the cylinder and the 'cpt' volume of the pulse tube. The losses are evaluated to obtain the actual cooling capacity and net work done. Assumptions are made to simplify the mathematical model of the OPTR.

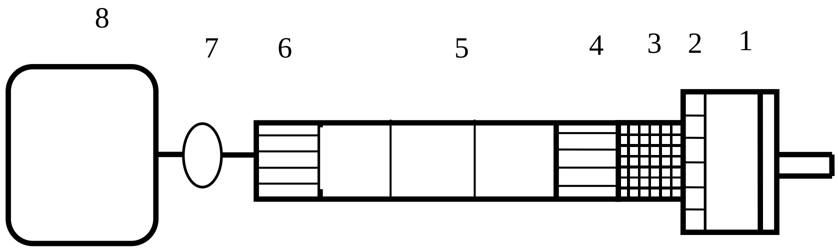

Figure 1: Schematic of an Orifice type Pulse Tube Refrigerator 1, Compressor; 2, After cooler; 3, Regenerator; 4, Cold Heat Exchanger; 5, Pulse Tube; 6, Hot Heat Exchanger; 7, Orifice; 8, Reservoir

Regenerator Model

The conflicting requirements for regeneration in an OPTR have been satisfied by selecting appropriate mesh material and size. The analytical model in the open form, based on finite difference technique of solution is classified into single and higher order model depending on the assumptions employed. In the present work, a higher order model is developed which accounts for the interaction of the regenerator and refrigerator processes. The processes include the temperature dependent properties of the matrix and working fluid, pressure drop, pressure variation *etc.*. Equations are complex when all the parameters of the equations are considered. To obtain solutions the equations are simplified using suitable assumptions [8]. The partial differential equations are nondimensionalized, discretised and solved by the method of Heggs [8] to predict the temperature distribution of working fluid and matrix along the length of the regenerator. The equations are coded in a QuickBasic computer program to simulate the regenerator as described by the authors [1] and integrated with the PTR program.

The performance depends on material, geometric and operating parameters of the system. Selecting the dimension of the pulse tube for a given compressor, regenerator geometry, charge pressure and cold end temperature operating at a particular frequency, the OPTR is designed.

Results and Discussion

Using the computer code developed to simulate an OPTR, performances are predicted for given input parameters. Configurations taken from published literature and a developed experimental setup is input into the simulation program to predict its performance. The predicted results are compared with actual experimental values for validation. The refrigerating effect, work done and COP along with velocity, mass flow and losses for a given cold heat exchanger temperature are investigated for sinusoidal pressure variations from a reciprocating compressor. The Parametric variation analysis of OPTR is carried out to analyse and identify the geometric and operating factors affecting its performance. Conceptual improvements are investigated for the optimisation of regenerator and thereby increasing its performance.

Performances are predicted by the simulation program for configurations obtained from published literature and experimental test setup. The refrigerating effect, work done and COP for a given cold heat exchanger temperature are compared with actual experimental values of the given configuration of Kral [10] for validation. An OPTR is designed and developed for Water Cooler applications. Experimental investigations are carried out, varying the operational parameter viz. frequency of compressor, orifice opening, charge pressure etc. The optimal performance of the PTR at a given charge pressure is obtained by adjusting orifice opening and varying frequency of compressor operation for minimum cold heat exchanger temperature. The specification of Kral's [10] OPTR is input into the computer program and its performance predicted.

Kral's OPTR

One cycle of rotation of the crank shaft is split into 32 equal intervals of 15^o crank angle. The pressures are calculated for the complete cycle. The pressure at the last interval, 360^o should be equal to the pressure at the start of the cycle within a tolerance limit of 0.15 bar. If not, the calculation of pressure restarts with a new value of initial pressure and temperature. The mean pressure is calculated as 16.341 bar for the cold heat exchanger temperature (T_{CHX}) of 90 K. The actual value of MR is calculated as 0.1362, the ratio of P_{avg} (16.5 bar) to P_{mean} which is the correct fraction of MR in the system. With this new value of MR all the pressure values calculated from equation 8 for each interval have to be multiplied so that P_{mean} matches P_{avg}. The total mass of gas in the OPTR is calculated to be 4.79345 grams.

Performance Parameters: The refrigeration effect is due to the working fluid displacement in the cold part of pulse tube 'cpt' in a given cycle. The ideal work input and refrigerating effect are 227.7 W and 2.9 W respectively for a T_{CHX} of 90 K. The losses in the OPTR system are calculated to obtain the actual refrigeration effect and net work done respectively. The performance is predicted for each T_{CHX} and is compared with experimental results of Kral [9]. At 90 K the simulation program predicts a refrigerating effect of 1.226 W. In contrast, at 274 K the refrigerating effect predicted by the program is 12.911 W, the actual experimental load being 11.57 W, a difference of 10.3%.

Experimentation

An experimental setup of OPTR is designed for domestic refrigeration application and developed using an open type reciprocating compressor [12]. Nitrogen is used as the working fluid. The configuration of experimental set up is listed in Table 1 and its overview is shown in Figure 3. Another OPTR is later assembled and set up on the second cylinder. The experimentation on a single OPTR is discussed. The measuring instruments used are:- Variac, Temperature Indicator for 'K' Type Thermocouple, Multimeter, Tachometer, amd Variable Frequency Drive, L&T make, 2 hp, 3 phase, 415 V. Performance of the setup is evaluated for varying operational parameters, for the purpose of validating the computer program. The refrigerating effect obtained is considerable in the 13^oC cold heat exchanger temperature. Hence, performance is predicted by the simulation program for a temperature of 13^{o}C T_{CHX} as shown in Figure 2.

Experimental investigations are carried out varying the operational parameters like frequency of compressor, orifice opening, charge pressure *etc*.. Comparisons between measurements and predictions of the simulation program are done at the optimum setting of the orifice, determined by experiments. The experimentation procedure and the performance results are highlighted. For a given charge pressure into the system, the rpm of motor is varied. At a particular speed of the motor the orifice valve is adjusted to get the lowest possible minimum temperature. On applying load through the heater wire, as the T_{CHX} rises, the orifice valve is adjusted again for drop in temperature. This process is continued for increasing load until the cold heat exchanger temperature reaches ambient temperature. The experiments are conducted with the compressor driven by a 1 hp motor. At higher charge pressures of 6,8 and 10 kg/cm2 the motor current increased above 2.1 A, the rated current indicating the inability of the 1 hp motor to cater to higher loads. A 2 hp motor could drive the compressor only at charge pressures of 9 kg/cm^2 and below. Hence the experiments were conducted at 9 kg/cm^2. At higher compressor speeds the performance of the PTR deteriorated. On disassembling, it was observed that oil was present in the PTR regenerator mesh.

Experimental Results

The experimental results obtained with the apparatus are analysed. The optimal performance of the PTR is obtained at a particular cold heat exchanger temperature by adjusting the orifice opening and frequency of compressor rotation. The performance from experimental results like capacity and COP at a given cold heat exchanger temperature are compared with their corresponding calculated parameters from the simulation program. The cooling performance using a pulse tube of 12 mm diameter is given in Table 2 for varying compressor speed.

The experimental result analysis is as follows:-

a) A temperature of -45^{o}C in the cold heat exchanger is obtained at an average pressure of 4 bar, metering valve opening of 0.35 m and a frequency of 21 Hz..
b) The optimum orifice opening of 0.65 mm diameter, 6.5 Hz. frequency of compressor and a charge pressure of 9 kg/cm^2 gives the maximum refrigeration effect of 18.06 W. When another OPTR is developed and fitted on the second cylinder a maximum refrigerating effect of 30.13 W is obtained for a net power input of 497 W, the COP being 0.06.
c) Inability of 1 hp motor to drive compressor to run the system above charge pressures of 6 kg/cm^2. At 9 bar average pressure the current rating of the motor increased to 2.7 A, leading to its heating up.
d) Oil ingress into OPTR above 25 Hz. leading to degradation in capacity.

Table 1: Configuration of OPTR Experimental Setup

Components	**Details**
Refrigeration Compressor	78.5 cm3 , Open type 0.5 H.P. 2 cylinders inline
Motor	Crompton Greaves make 1 HP, 400 V, 2.1 A , 1400 rpm
Metering Valve	Swagelok SS-6L-MH, 3.25 mm diameter with Vernier handle
Variable Frequency Drive	Yaskawa-L&T make, 2HP
Reservoir	Refrigerant Container, 948 cc
Pre Cooler	φ 1.3 cm, 6.5 cm length filled with 80 mesh SS screen disks, Water Cooled, Tube in Tube type Heat Exchanger
Regenerator	φ 1.73 cm, 28 cm length filled with 80 mesh SS screen disks
Cold Heat Exchanger	φ 1.8 cm, 5..38 cm length filled with 30 mesh Copper screen disks
Pulse Tube	S.S., φ 1.2 cm, 35 cm length
Hot Heat Exchanger	φ 1.3 cm, 15 cm length filled with 30 mesh SS screen disks, water cooled, Tube in Tube type
Temperature Indicator	'K' Type Thermocouple
Working Fluid	Nitrogen

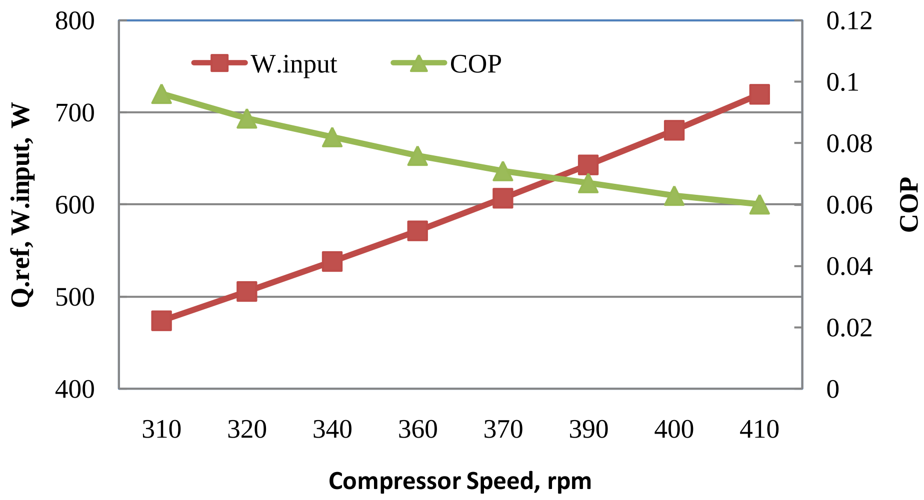

Figure 2: Theoretical Performance of an OPTR

Charge pressure 9 kg/cm^2 (8.82 bar) and T_{CHX} 13^{o}C

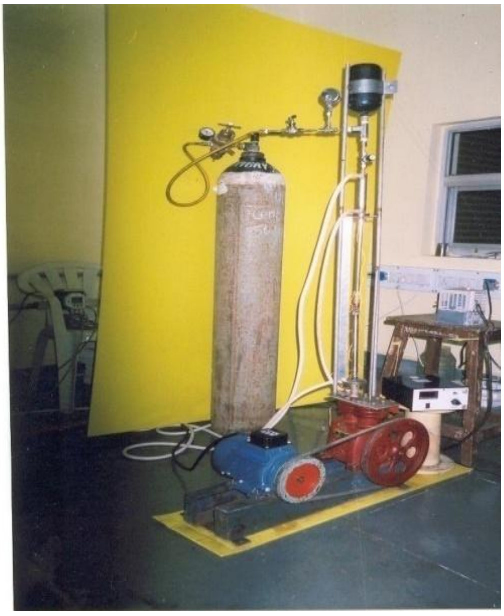

Figure 3: Experimental Setup of OPTR

e) Mass flow rate of water through the Precooler and hot heat exchanger at 0.8 kg/s.
f) Cover plate temperature 84^0C at 9 kg/cm^2. Water cooling provided to reduce heat of compression.
g) The system operates continuously for 8 hours at 4 kg/cm^2 without degradation in performance.

At 390 rpm of the compressor speed, the load on PTR was 18.056 W, whereas the theoretical prediction from the simulation program was 22.02 W, a difference of 18%. The loss due to heat exchanger ineffectiveness of the precooler and hot heat exchanger has not been quantified, while calculating the refrigerating effect. The deviation will thus be lower than the estimated value of 18%. The simulation program is validated to predict the performance of an Orifice Pulse Tube Refrigerator. Hence, the program developed can be used in optimizing existing systems and developing new OPTRs for a given application.

Conclusions

A simulation program which can be used to design and optimize an OPTR for applications in domestic refrigeration is developed by the authors. At 274 K the refrigerating effect predicted by the program for a configuration developed by Kral [10], is 12.911 W, whereas the actual experimental load is 11.57 W, a difference of 10.3%. A maximum refrigerating effect of 30 W is obtained for a net power input of 497 W, the COP being 0.06 at 390 rpm of compressor speed and charge pressure of 9 kg/cm^2. The system operates continuously at 4 kg/cm^2 charge pressure without degradation in performance. A Stirling type OPTR coupled to a 1 ton open type refrigeration compressor using Nitrogen as the working fluid operating at 4 kg/cm^2 can be developed for a 10 lts/hr water cooler. A low cost oil free open type reciprocating compressor will make the Pulse Tube Water Cooler a reality.

Table 2: Experimental results of an OPTR with charge pressure of 9 kg/cm^2 (8.82 bar) and T_{che} 13^{0}C

N_{comp}	Q_{ref}, W	W_{input},W	Base load, W	W_{net}, W	COP
310	14.04	636.9	250.5	386.4	0.068
320	14.38	666	262.5	403.5	0.068
340	13.21	712.8	276	436.8	0.063
360	15.34	675.9	288	387.8	0.08
370	16.64	755.7	300	455.7	0.067
390	18.06	809.2	312	497.2	0.06
400	13.08	896.4	324	572.4	0.035
410	9.48	921.6	336	585.6	0.025

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