

Article

Thermodynamic Analysis of a Waste Heat Driven Vuilleumier Cycle Heat Pump

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Abstract: A Vuilleumier (VM) cycle heat pump is a closed gas cycle driven by heat energy. It has the highest performance among all known heat driven technologies. In this paper, two thermodynamic analyses, including energy and exergy analysis, are carried out to evaluate the application of a VM cycle heat pump for waste heat utilization. For a prototype VM cycle heat pump, equations for theoretical and actual cycles are established. Under the given conditions, the exergy efficiency for the theoretical cycle is 0.23 compared to 0.15 for the actual cycle. This is due to losses taking place in the actual cycle. Reheat losses and flow friction losses account for almost 83% of the total losses. Investigation of the effect of heat source temperature, cycle pressure and speed on the exergy efficiency indicate that the low temperature waste heat is a suitable heat source for a VM cycle heat pump. The selected cycle pressure should be higher than 100 MPa, and 200–300 rpm is the optimum speed.

Keywords: VM cycle heat pump; thermodynamic calculation; exergy efficiency analysis

1. Introduction

The energy crisis and environmental pollution are two of the world's major problems [1,2]. Utilization of waste heat is a better way to combat these problems, for this method not only supplies more energy, but also simultaneously reduces the environmental pollution [3]. The amount of waste heat is huge. For example, the energy consumption in China during 2013 was 3.75 billion tons coal equivalent, and the

overall energy conversion efficiency is about 72.3% [4], which means the waste heat is over 1 billion tons coal equivalent. However, how to access and use this waste energy is a tough problem.

Heat pump technology is recognized as the most efficient way to utilize the waste energy. The coefficient of performance (COP) of an electric or mechanical driven heat pump is always greater than 1. In some cases, such as when high temperature waste heat sources are available, heat driven heat pumps will be more preferable. Absorption, adsorption and steam ejection are the most common types [5]. In general, the COP of a double effect absorption heat pump would be greater than 1, up to 1.56. The COP of adsorption is typically under 0.7, and even lower for steam ejection technology. The Vuilleumier (VM) cycle is also a heat activated closed gas cycle. It can utilize thermal energy, such as solar energy, isotope radioactivity, waste heat, *etc.* Experimental COP can reach 1.61, which is the highest among heat driven heat pump technology [6].

The VM cycle was patented by Rudolph Vuilleumier in 1918 [7]. This cycle has many advantages, such as fewer moving parts, a closed cycle, making it easier to control the pollution, potential higher COP, lower noise, lower mechanical friction and usefulness for distributed energy systems with small power output [8–12]. This cycle can also be applied in the cryogenic field [13,14]. Unfortunately, disadvantages always exist along with advantages. The cylinder volumes are not isothermal but nearly adiabatic. Even if the heat exchangers and the regenerators are perfect, the COP will be lower than for the Carnot cycle because the mean temperature in the cylinder volumes differs from the temperature in the heat exchanger brine. For a cooling cycle based on the VM cycle, the temperature difference between the two cylinder volumes is often smaller than the change of temperature in the cylinder volumes themselves, and the COP for the cooling cycle is thus seriously reduced. Another disadvantage is the specific output is low. For a desirable heat output the machine must be large, and therefore rather expensive.

The first application of the VM cycle heat pump in the room temperature field was in 1989 by Carlsen. He designed a natural gas driven VM cycle heat pump with helium as refrigerant for residence heating. The heating capacity output is 7.5 kW, and the COP reaches 1.62 [15]. Later he developed a 20 kW gas fired VM cycle heat pump [16]. Kuehl transformed the crankshaft driven VM heat pump into the free piston driven type [17,18]. Lee analyzed the influence of structure on the performance of VM heat pump, and the results show that the structure of the regenerator has a significant effect on the performance [19]. Finkelstein [20], Sekiya [21,22] and Kawajiri [23] have presented analyses of system performance. Xie [24] carried out research on the influence of binary mixtures on the regenerator of a VM heat pump.

Up to now, no work has related the second law of thermodynamics to a VM cycle heat pump. To comprehensively understand the VM cycle heat pump, an exergy analysis is carried out along with an energy analysis in this paper to evaluate the application of a VM cycle heat pump in waste heat utilization.

2. VM Cycle Heat Pump

2.1. Configuration

Figure 1 shows the construction schematic of a waste heat driven VM cycle heat pump. The basic VM cycle heat pump contains two cylinders, namely a cold cylinder and a hot cylinder. Each cylinder

has a displacer which performs reciprocating motion inside the cylinder. A seal device is used between the displacer and the cylinders for anti-leakage purposes. The hot and cold displacers are driven by a crankcase with a connecting rod.

As shown in Figure 1, the space between the hot displacer and the top of the hot cylinder is called hot space, while the cold space is between the cold displacer and the top of the cold cylinder. The space between the cold displacer, hot displacer and the crankshaft is called warm space. A cold regenerator is placed between the cold space and warm space, while a hot regenerator is between the warm space and hot space. The regenerators work as a heat storage system when the working fluid flows between spaces.

The input high temperature waste heat is absorbed by the hot heat exchanger, mixed with low temperature heat from the surroundings which is absorbed by cold heat exchanger, providing heating from the warm heat exchanger for domestic purposes.

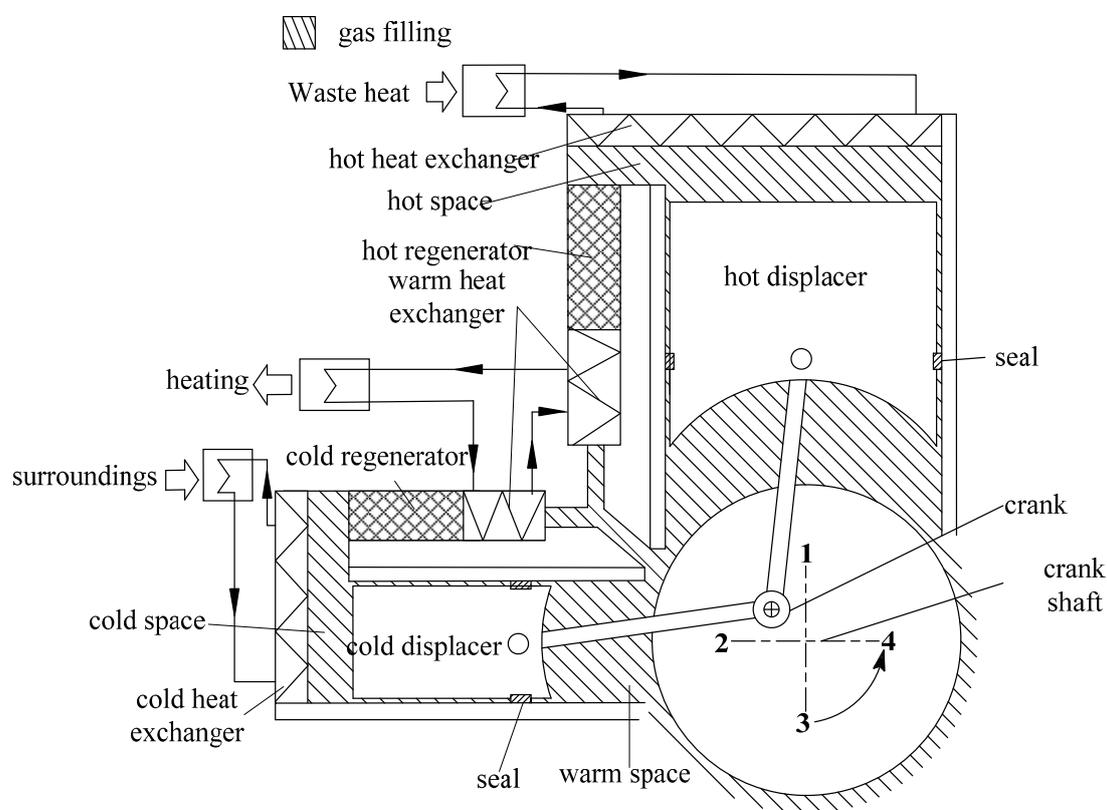


Figure 1. Schematic of a Vuilleumier cycle heat pump.

2.2. Working Process

The working process for a VM cycle heat pump is shown in Figure 2. The positions of the starting points are shown in Figure 1.

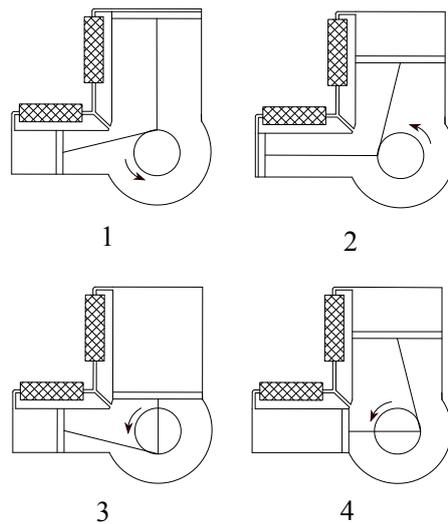


Figure 2. Working processes of a Vuilleumier cycle heat pump.

The cycle starts at point 1, and the whole process is illustrated as follows:

Process 1–2: the crankcase rotates counterclockwise, making the crankshaft push the hot displacer to the bottom of the hot space, while the cold displacer moves to the top of the cold space. During this process, the hot space volume increases, and the cold space volume decreases. As the hot displacer moves to the bottom, the working fluid from the warm space is heated to close to the hot space temperature (T_h) by filling materials when going through the hot regenerator, and then the working fluid goes into the hot space. The working fluid in the cold space is pushed by the cold displacer to the warm space, where it is heated to near the warm space temperature (T_a) by the filling material in the cold regenerator. Both the temperature and the pressure of the VM cycle heat pump increase rapidly. Output heat is supplied by the warm heat exchanger in the warm space.

Process 2–3: the crank moves from left to the bottom. The hot displacer still goes to the bottom of hot space. Meanwhile the cold displacer leaves the top to the bottom of cold space. The volumes of both are increasing. During this process, the working fluid in the warm space is cooled to nearly the cold space temperature (T_c) by the filling material in the cold regenerator, and then enters the cold space. Gas in the hot space absorbs heat from high temperature waste heat through the hot heat exchanger, while the cold space is absorbing heat from the low temperature surroundings using the cold heat exchanger.

Process 3–4: the crank rotates from bottom to right. The hot displacer leaves the bottom of the hot space to the top, and the cold displacer keeps moving to the bottom of the cold space. The hot space volume decreases, while the cold space volume keeps increasing. During this process, the working fluid in the hot space is pushed into the warm space by the hot displacer and cooled to nearly the warm space temperature (T_a) by the filling materials when crossing the hot regenerator. The gas temperature decreases, and the pressure also drops.

Process 4–1: the crank moves from right to the top. The hot displacer moves to the top of the hot space, and the cold displacer leaves the bottom to the top of cold space. The hot space volume decreases to nearly 0, and the cold space volume decreases continuously. During this process, gas in the hot space is cooled to nearly the warm space temperature (T_a) by the filling in the hot regenerator, then goes into warm space, with the pressure drop. A part of the gas in the cold space is heated to nearly the warm

space temperature by the filling in the cold regenerator when it passes through the cold heat exchanger, and then returns to the warm space.

3. Thermodynamic Analysis

Simultaneous calculation and separate calculation are the two main methods to perform the thermodynamic analysis for a VM cycle heat pump [25]. In this paper, the separate calculation method based on an isothermal model is chosen. For the theoretical cycle, it assumes the following:

- (1) Processes occurring in all spaces are isothermal, *i.e.* the temperature of the compression or the expansion process is stable.
- (2) No losses occur in all the processes within the cycle, which means there is no heat transfer or flow resistance.
- (3) The working fluid is considered as ideal gas.

3.1. Working Fluid

The common working fluids adopted by the VM cycle heat pump are helium, hydrogen and nitrogen. As the VM cycle heat pump is a closed gas cycle, these working fluids are always in gas phase without any phase change. The properties of these working gases deviate from those of the ideal gas, so the different working gases adopted in a VM cycle heat pump have different system performance.

Generally hydrogen is the best working gas, but hydrogen is too active and it can easily explode when leaks occurs. Figure 3 shows that the differences in the thermodynamic performance of a VM cycle heat pump using hydrogen or helium as the working gas are small, and greater than when nitrogen is used as working gas. For both safety and performance concerns, helium is selected as the working gas for the VM cycle heat pump. All calculations carried out in this paper are based on helium as refrigerant. Some properties of helium are listed in Table 1.

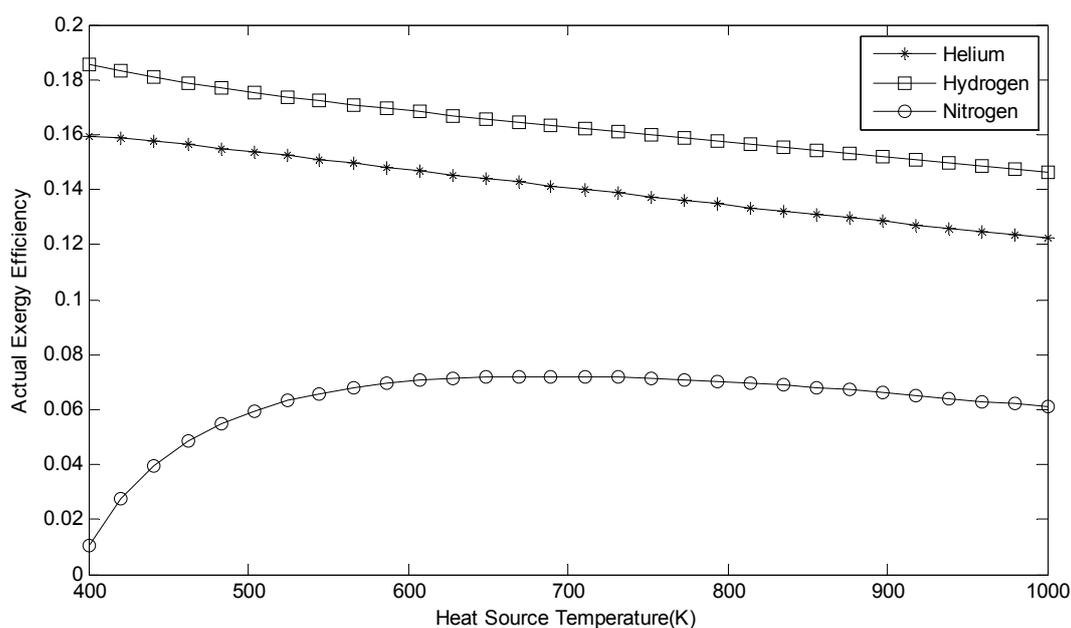


Figure 3. The influence of working fluids on exergy efficiency.

Table 1. Some properties of helium.

Parameter	Symbol	Value
Working fluid	<i>He</i>	—
Universal gas constant	<i>R</i>	8.314 J/(K·mol)
Specific heat	<i>C_P</i>	5.187×10 ³ J/(kg·K)
Thermal conductivity	<i>λ</i>	0.306W/(m·K)

3.2. Calculation Parameters

Table 2 gives the parameters needed for calculation, including constructional parameters and working conditions of the VM cycle heat pump.

Table 2. Primary calculation data.

Parameter	Symbol or Instruction	Value
Cylinder bore	<i>D_{co}</i>	0.0699 m
Stroke	<i>Z</i>	0.0312 m
Space phase angle	<i>Φ</i>	90°
Displacer length	<i>L</i>	0.04359 m
Displacer radial clearance	<i>δ</i>	0.00015 m
Regenerator diameter	<i>D_R</i>	0.0226 m
Regenerator length	<i>L_R</i>	0.0226 m
Regenerator material	Stainless steel wire	—
Hot space temperature	<i>T_h</i>	553 K
Warm space temperature	<i>T_a</i>	338 K
Cold space temperature	<i>T_{co}</i>	263 K
Surrounding temperature	<i>T₀</i>	273 K
Average temperature	$T_{av} = \frac{T_h + T_{co}}{2}$	408 K
Average pressure	<i>P_{av}</i>	10.0×10 ⁶ Pa
Drive system	Single crank, Double-acting pistons drive	—
Rotational speed	<i>n</i>	600 rpm = 10 Hz
Volume ratio	<i>ω</i>	10

The heating output will be supplied as a radiator system or floor heating system. Radiator systems are more common, but the literature shows that floor heating systems are more comfortable. Floor heating systems have higher Predicted Mean Vote (PMV) and low vertical temperature differences. Floor heating systems also have energy saving advantages, as the system supply temperature is normally under 60 °C (typically it is 55 °C). The system return temperature is set at 35 °C.

3.3. Equations for the Energy and Exergy Analyses

All the equations used in this paper are referred to literature [8].

Heat absorbed by the cold space Q_{co} , referred to as the refrigerating capacity, can be defined as:

$$Q_{co} = P_{av} V_{co} \frac{\pi n \delta \sin \theta}{60(1 + \sqrt{1 - \delta^2})} \tag{1}$$

where p_{av} is the average space pressure; V_{co} is the max volume of the cold space; n denotes the speed of the crank rotation; δ is the pressure parameter and θ is the phase angle.

Heat transferred from the waste heat in hot space Q_h is:

$$Q_h = Q_{co} \frac{\tau_{co} - 1}{1 - \tau_h} \tag{2}$$

where τ_{co} denotes the temperature ratio at the cold side, $\tau_{co} = T_a/T_c$; τ_h is the temperature ratio at the hot side, $\tau_h = T_a/T_h$.

Q_a is the heat output for heating purposes from the warm space:

$$Q_a = Q_{co} + Q_h \tag{3}$$

For an actual VM cycle heat pump, there exist lots of irreversible losses, such as reheat losses, flow friction losses, shuttle losses, pumping losses, heat conduction losses, leakage losses and so on. Reheat loss ΔQ_R represents the heat transfer losses in the heat exchanger of the regenerator caused by temperature differences, wall effects, temperature swings of materials, etc.:

$$\Delta Q_R = c_p M_0 (1 - \eta_R) (T_h - T_c) \tag{4}$$

where c_p is the specific heat at constant pressure; M_0 is the average mass flux through the regenerator; η_R is the regenerator efficiency, typically $\eta_R = 0.98$.

Flow friction loss ΔQ_f refers to the pressure losses when the working gas passes through the regenerator, heat exchanger and ducts:

$$\Delta Q_f = \Delta P \frac{V_{co} n}{30} \tag{5}$$

where ΔP is the pressure drop.

In a VM cycle heat pump, displacers have the same axial temperature distribution as the cylinders. Due to the reciprocating motion of the displacers, there exists a temperature difference between the displacers and cylinders. This heat transfer loss is called shuttle loss ΔQ_{sh} :

$$\Delta Q_{sh} = F_s \frac{\lambda \pi D_{co} Z^2 (T_h - T_c)}{5.4 \sigma L} \tag{6}$$

where F_s is a correction factor, which generally has a value of 0.52–0.73; λ is the thermal conductivity, D_{co} is the diameter of the cold cylinder; Z is the displacer stroke; δ is the radial clearance between the displacer and cold cylinder; L is the displacer length.

Because of the temperature difference, the working gas entering each space will result in an extra heating load. This is called pumping loss ΔQ_p :

$$\Delta Q_p = \frac{2(\pi D_{co})^{0.6} L (P_{max} - P_{min})^{1.6} n^{1.6} c_p^{1.6} (T_h - T_c) \sigma^{2.6}}{1.5 Z_p (R')^{1.6} \lambda^{0.6} [(T_h + T_c)/2]^{1.6}} \tag{7}$$

where P_{max} is the maximum pressure during the cycle; P_{min} is the minimum pressure during the cycle; Z_p is the gas compressibility factor.

In an actual VM cycle heat pump, the length of the rod of displacers is short, resulting in heat conduction losses. The heat conduction loss ΔQ_d is the sum of the wall conduction loss ΔQ_{dw} and the filling material conduction loss ΔQ_{dm} :

$$\Delta Q_d = \Delta Q_{dw} + \Delta Q_{dm} \quad (8)$$

The wall conduction loss ΔQ_{dw} is:

$$\Delta Q_{dw} = \frac{A}{L} \left[\int_4^{T_a} \lambda(T) dT - \int_4^{T_c} \lambda(T) dT \right] \quad (9)$$

where A is the cross-sectional area.

Filling material conduction loss ΔQ_{dm} can be expressed as:

$$\Delta Q_{dm} = \frac{A}{L_R} \lambda_m (T_h - T_c) \quad (10)$$

There is also a temperature difference between the VM cycle heat pump and surroundings. This heat loss is called heat dissipation loss, ΔQ_l . We can take account for 5% of the Q_{co} , so the total losses of a VM cycle heat pump are:

$$\Delta Q = \Delta Q_R + \Delta Q_f + \Delta Q_{sh} + \Delta Q_p + \Delta Q_d + \Delta Q_l \quad (11)$$

The theoretical and practical aspects of thermodynamics are most relevant to energy and exergy analysis. Exergy analysis provides an alternative and illuminating means of assessing and comparing processes and systems rationally and meaningfully. Exergy analysis can assist in improving and optimizing designs.

The specific exergy flow of the working gas or water is evaluated as follows:

$$ex_{r,w} = (h - h_0) - T_0(s - s_0) \quad (12)$$

where h is the enthalpy; s is the entropy; and the subscript zero indicates properties at the reference (dead) state (i.e., at P_0 and T_0).

Here we suppose that the temperature T of the waste heat is constant, then the thermal exergy of waste heat at heat flow Q transferred to the VM cycle heat pump is:

$$Ex_Q = \left(1 - \frac{T_0}{T}\right)Q \quad (13)$$

The exergy efficiency COP_{ex} [26] is generally expressed as the ratio of total exergy output to total exergy input and is written as follows:

$$COP_{ex} = \frac{Ex_{output}}{Ex_{input}} \quad (14)$$

where “output” refers to “net output” or “product” or “desired value”, and “input” refers to “driving input” or “fuel”. For a VM cycle heat pump, the output is the heating, with T_{out} at 55 °C and T_{in} at 35 °C, the input is the waste heat and/or the heat from the surrounding different than the dead state. Exergy efficiency for a VM cycle heat pump can then be written as follows:

$$COP_{ex} = \frac{E_{x,out} - E_{x,in}}{E_{x,h} + E_{x,c}} = \frac{Q_a \left(\left(1 - \frac{T_0}{T_{out}} \right) - \left(1 - \frac{T_0}{T_{in}} \right) \right)}{Q_h \left(1 - \frac{T_0}{T_h} \right) + Q_{co} \left(\frac{T_0}{T_c} - 1 \right)} \tag{15}$$

3.4. Calculation Results

Table 3 shows the calculation results for the theoretical VM cycle heat pump. The assumptions of the theoretical VM cycle heat pump are as stated before, which include no irreversible losses. With the parameters in Table 3, the results show the theoretical exergy efficiency is 0.23.

Table 3. Theoretical calculation results.

Item	Symbol	Value
Stroke volume of cold space	V_{co}	$119.73 \times 10^{-6} \text{ m}^3$
Temperature ratio	τ	2.1
Pressure phase angle	θ	85.8°
Temperature ratio at cold side	τ_{co}	1.3
Temperature ratio at hot side	τ_h	0.6
Pressure factor	σ	0.2
Pressure ratio	P_{max}/P_{min}	1.5
Maximum pressure	P_{max}	$12.4 \times 10^6 \text{ Pa}$
Minimum pressure	P_{min}	$8.1 \times 10^6 \text{ Pa}$
Theoretical cooling of cold space	Q_{co}	4020.1 W
Theoretical heat consumption of hot space	Q_h	2948.7 W
Theoretical heat rejection of warm space	Q_a	6968.8 W
Theoretical exergy efficiency	$COP_{ex,i}$	0.23

Table 4 is the result of the actual process. The actual exergy efficiency is 0.15, much lower than the theoretical value of 0.23. The reason is the losses generated during the actual process. Among these losses, reheat loss and flow friction loss account for a large proportion of the total loss, almost 83%.

Table 4. Calculation results of the actual process.

Subject	Symbol	Value
Reheat loss	ΔQ_R	1317.9 W
Flow friction loss	ΔQ_f	1196.6 W
Shuttle loss	ΔQ_{sh}	322.4 W
Pumping loss	ΔQ_p	0.8 W
Wall heat conduction loss	ΔQ_{dw}	6.7 W
Filling heat conduction loss	ΔQ_{dm}	1.7 W
Heat conduction loss	ΔQ_d	8.4 W
Leakage loss	ΔQ_l	201.0 W
Total loss	ΔQ	3046.9 W

Table 4. Cont.

Subject	Symbol	Value
Actual cooling	$Q_{co,ac}$	1497.3 W
Actual heat rejection	$Q_{a,ac}$	4769.1 W
Actual heat consumption	$Q_{h,ac}$	3271.8 W
Actual exergy efficiency	$COP_{ex,e}$	0.15

3.5. The Influence of Parameters on Exergy Efficiency

The theoretical and actual exergy efficiencies calculated above are based on the given parameters. Some parameters may affect the exergy efficiency greatly, while others only have a slight effect. The following is the analysis result of the effect of some parameters on the exergy efficiency for the calculated prototype VM cycle heat pump.

Figure 4 shows the influence of heat source temperature on the exergy efficiency. As the temperature of the waste heat increases, the theoretical exergy efficiency decreases, while the actual first increases then decreases, reaching a maximum at the point of 370K. This means that we do not need to have much higher temperature waste heat since lower heat source temperature results in higher exergy efficiency. For most waste heat the temperature is under 600K, so a VM cycle heat pump can be used for waste heat utilization.

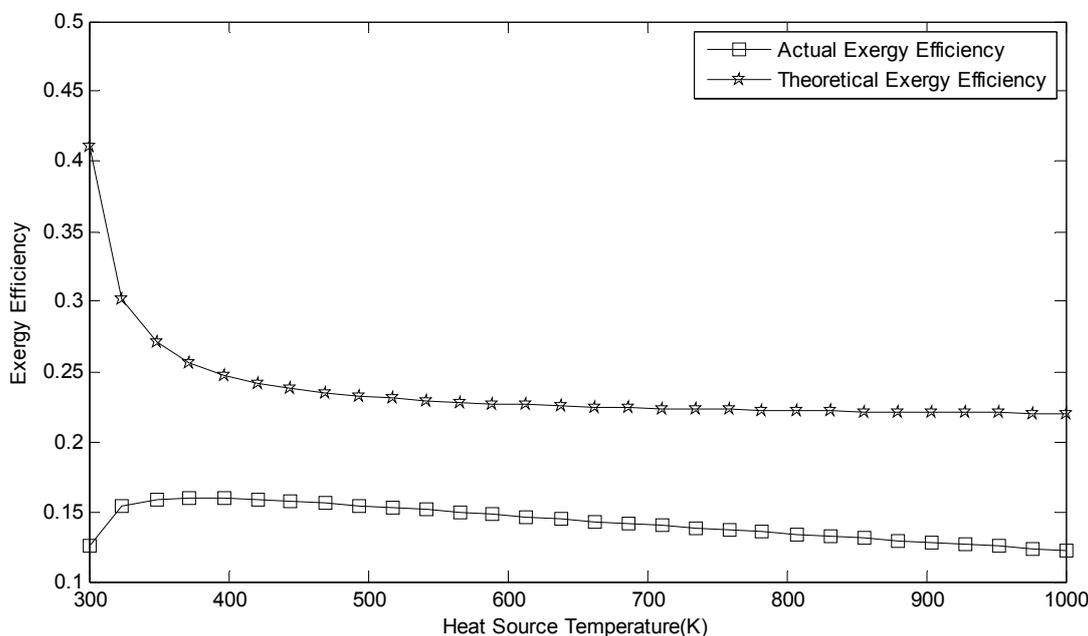


Figure 4. The influence of heat source temperature on exergy efficiency.

The influence of cycle pressure on actual exergy efficiency is shown in Figure 5. As the pressure increases, the actual exergy efficiency also increases. After the cycle pressure reaches 100MPa, the exergy efficiency becomes stable. For most cases, the optimum pressure is hard to reach, so it can be treated as a reference to obtain higher performance.

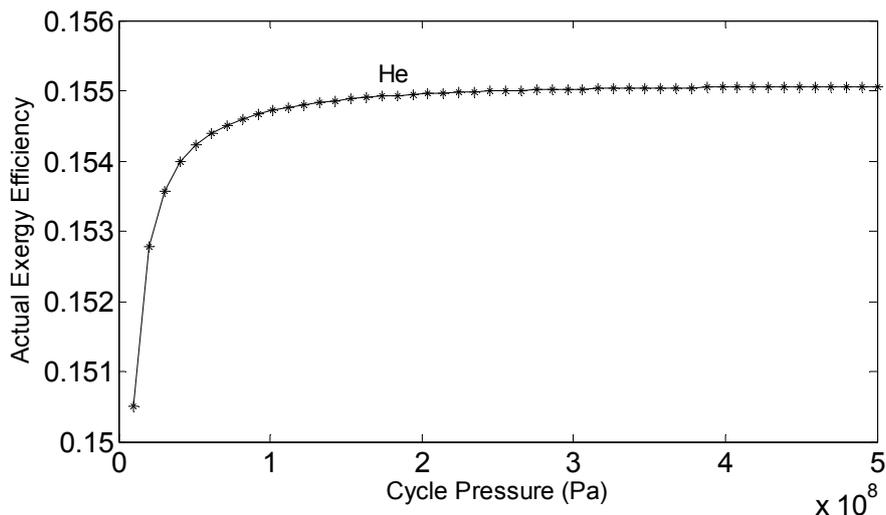


Figure 5. The influence of cycle pressure on the actual exergy efficiency.

Figure 6 presents the influence of speed and heat source temperature on actual exergy efficiency. A faster speed leads to lower exergy efficiency, while the amplitude of the exergy efficiency decrease increases.

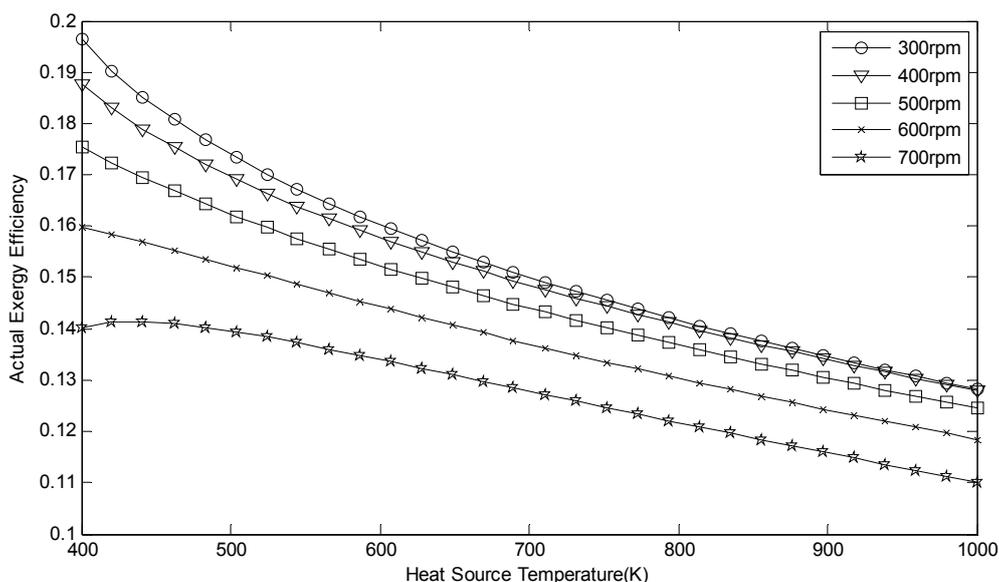


Figure 6. The influence of speed and heat source temperature on actual exergy efficiency.

4. Conclusions

In this paper, analyses of the theoretical and actual cycle of a VM heat pump are presented. Besides, the influence of some parameters on exergy efficiency is given. The conclusions are as follows:

- (1) According to the calculation results, the theoretical and actual exergy efficiency are 0.23 and 0.15, separately. Reheat losses and flow friction losses account for almost 83% of the total loss.
- (2) As the temperature of waste heat increases, the theoretical exergy efficiency decreases, while the actual one first increases then decreases, reaching a maximum at the point of 370K. Low temperature

waste heat under 600K temperature is more suitable as the driving heat source of a VM cycle heat pump.

- (3) The cycle pressure and speed also have great influence on the exergy efficiency. The selected cycle pressure should be higher than 100MPa, and 200–300rpm is optimum for speed.

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Author Contributions

Yingbai Xie proposed and designed the research; Xuejie Sun performed the calculations. Both authors jointly worked on deriving the results and wrote the paper. Both authors have read and approved the final manuscript.

Nomenclature

Symbol	Definition	Symbol	Definition
D	Diameter (m)	M	Mass flux (kg s^{-1})
Z	Stroke (m)	ΔP	Pressure drop (Pa)
Φ	Volume phase angle ($^{\circ}$)	F_s	Correction factor
L	Length (m)	Δ	Heat loss (W)
δ	Clearance (m)	Z_p	Gas compressibility factor
T	Temperature (K)	COP_{ex}	Exergy efficiency
P	Pressure (Pa)		
n	Rotational speed (rpm)	<i>Subscripts</i>	
ω	Volume ratio	co	Cold space
He	Helium	h	Hot space
R	Universal gas constant ($\text{JK}^{-1}\text{mol}^{-1}$)	a	Warm space
C_p	Specific heat ($\text{Jkg}^{-1}\text{K}^{-1}$)	R	Regenerator
V	Volume (m^3)	av	Average
τ	Temperature ratio	max	Maximum
Q	Heat quality (W)	min	Minimum
θ	Pressure phase angle ($^{\circ}$)	m	Material
ψ	Porosity	f	Flow
D	Diameter (m)	sh	Shuttle
A	Area (m^2)	p	Pumping
a	Specific heat transfer area (m^2)	i	Ideal
COP	Coefficient of performance	e	Effective
r	Hydraulic radius (m)		
ρ	Density (kgm^{-3})		

Conflicts of Interest

The authors declare no conflict of interest.

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