A PROPOSED THEORETICAL STANDARD CYCLE FOR A RECIPROCATING STEAM ENGINE

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Abstract. *The reciprocating steam machine played an important role in the industrial revolution and it was widely used in many sectors of industry. But with the development of the internal combustion engines and steam turbines, this technology was virtually abandoned. Nowadays the interest in steam machine began to grow again, due to the growth of distributed generation of eletricity. This type of power generation usually uses the organic Rankine cycle (ORC) to generate low power, making the steam machine viable when compared to the steam turbines. In the literature, each author uses a different cycle to study or design his equipment, so it is interesting to study a single standard cycle for the reciprocating steam engines to serve as standard for comparing cycles already studied. The results show that there are maximum operational points, which depend in some conditions and constructive features.*

Keywords: *Steam engine, Organic Rankine Cycle (ORC), Standard cycle, Expansion machine*

1. INTRODUCTION

To develop a cycle for the reciprocating machine, it is necessary to understand and study their mode of operation. The reciprocating steam machine played an important role in the industrial revolution and was widely used in many sectors of industry in the previous centuries. In the beginning of the 20^{th} century, this technology was replaced by engines which operate according to Otto and Diesel cycles and, at the same time, steam turbines were conceived. In any case, steam machines were relegated to a second plane due its low efficiency and to a very low power generation to volume density ratio. During the 20^{th} century, the steam machine technology remained stagnant e without any practical use.

Recently the steam machine became again an interesting technology. This fact comes from the increasing interest in Organic Rankine Cycles (ORC), used to generate energy through low power systems installations, generally using solar energy as a heat source. The ORC are most used in places away from the large centers, where the electricity is more difficult to be obtained. Because of its use and constructive form, the powers involved are small and according to Prasad (1993), for low output powers (a few hundred kilowatts), steam motors are still superior to turbines in most respects. Thereby, the steam machine becomes a viable alternative to replace the turbine as the ORC equipment expansion. Literature on steam machine lacks of precise information of a standard cycle in a similar fashion as the Otto and Diesel cycles. So it is interesting to establish and study a standard cycle that can be used by anyone, without neglecting the various cycles already studied.

2. DEVELOPMENT

To develop a standard cycle for reciprocating steam machines, it is necessary to understand and study their mode of operation. The cycle used in earlier studies on steam engines is shown in Fig. 1 (Marino, 1948), and it is still used nowadays by some authors (Alanne et al., 2012). The cycle begins at the top dead center (1) with steam at high pressure and high temperature entering the cylinder and pushing the piston toward the bottom dead center. The intake valve closes at a point (2) along the movement of the piston, terminating the admission process and starting the expansion process. The steam expands until it reaches the condenser pressure at the bottom dead center (3). At this point, the exhaust valve opens instantaneously and the steam is expelled from the cylinder during the return stroke of the piston to top dead center (4). On reaching the top dead center, the intake valve opens instantaneously, increasing the internal pressure of the cylinder to the intake pressure and restarting the cycle. This cycle, as well as other cycles studied, is not exclusive of the steam engine, it can be used by any equipment operating cyclically by performing the expansion of a high pressure gas, such as compressed air motors.

2.1. Indicated diagrams

However in practical terms, it is not feasible to build a machine with a zero dead volume, because it would cause an impact between the piston and the cylinder head. This impact causes great stress on parts and a significant reduction in equipment life. For this reason it is necessary to have a non-zero dead volume, but for purpose of efficiency is interesting that this dead volume is minimized. Prasad (1993) and Baek et al. (2005) consider that the ideal cycle is the one shown in Fig. 1, but considering a non-zero dead volume.

Because of the dead volume, as the internal pressure of the cylinder is lower than the evaporator pressure at the point of the exhaust valve closing and opening the inlet valves (point 4), a portion of the high-pressure steam that enters in cylinder will fill the dead volume and increase the internal pressure until it reaches the pressure of the inlet steam. This portion of steam does not generate useful work, reducing the efficiency of the machine.

To solve this issue and make no part of steam be wasted in increasing the pressure of the dead volume, the pressure inside the cylinder must be equal to the inlet pressure when the intake valve opens. To make this possible, the exhaust valve must close before the piston reaches the TDC. Therefore, the range in which both valves are closed, the steam undergoes compression, increasing the pressure until it reaches the inlet pressure. Thus, when the intake valve opens, the internal pressure of the cylinder is the inlet pressure and no steam is used to fill the dead volume. This cycle is used by Trajkovic (2010) and the corresponding pressure-volume indicated diagram is shown in Fig. 2. If the expansion and compression of steam are isentropic, then there are two isobaric processes and two isentropic processes.

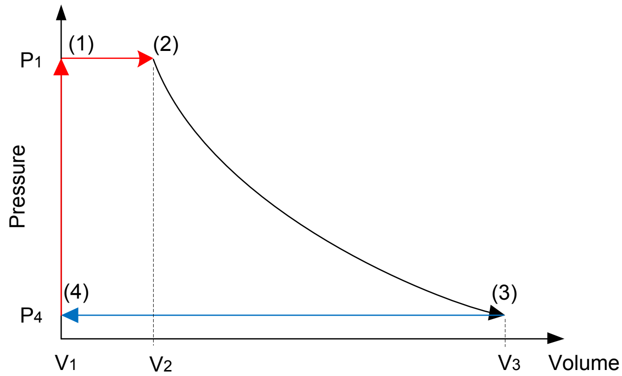

Figure 1. Pressure-volume indicated diagram for an ideal cycle for a reciprocating steam machine without dead volume (Marino (1948) and Alanne et al. (2012)).

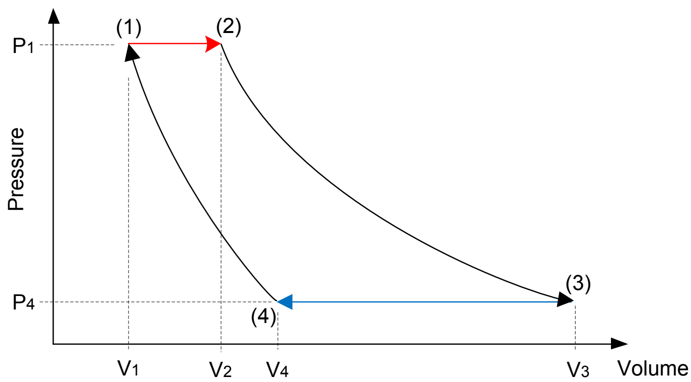

Figure 2. Pressure-volume indicated diagram for an ideal cycle with dead volume and vapor compression (Trajkovic (2010)).

Generally, due to technical limitations, it is not possible to achieve the complete expansion of steam (process 2-3 of Fig. 2) or the complete compression of steam until the inlet pressure (process 4-1 of Fig. 2). These limitations may be due to a variable inlet pressure, there may be a high inlet pressure, and the volume ratio is not enough to completely expand the steam, or in the end of expansion, the working gain is so small as to be smaller than the energy dissipated by the friction. These limitations lead to the incomplete expansion and/or compression of steam. Based on this, Prasad (1993) solve this issue in his simulations using the cycle shown in Fig. 3, where there is an incomplete expansion and an incomplete compression of the steam inside the cylinder. It makes the efficiency to become lower when compared with the efficiency of the cycle of Fig. 2.

Badami and Mura (2009) use a cycle with an incomplete expansion and compression, but they consider steam inlet along the piston stroke. This cycle is represented in Fig. 4 and has an isentropic efficiency lower than 1, but it is closer to the actual operation of the steam engines than the previous cycles.

Antonelli and Martorano (2012) e Basbous et al. (2012) use the indicated diagram of Fig. 5, which is the closest cycle to the actual operation of the steam engine. This cycle takes into account that the exhaust valve remains opened during a portion of the return stroke of the piston, and along this stroke it closes, causing the vapor to be compressed until the piston reaches the TDC. This cycle becomes the most generalized cycle, where all the other cycles become special cases of this one. By allowing a study of the points of opening and closing the valves and for being the nearest cycle of the real operating cycle of a steam engine, this cycle can become the starting point of any study on the working of steam engines.

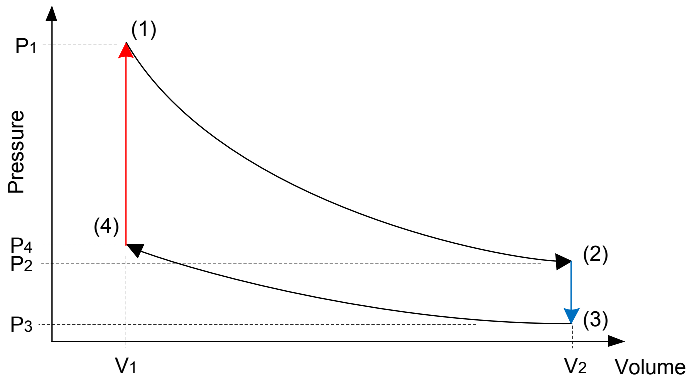

Figure 3. Pressure-volume indicated diagram with incomplete expansion and compression (Prasad, 1993).

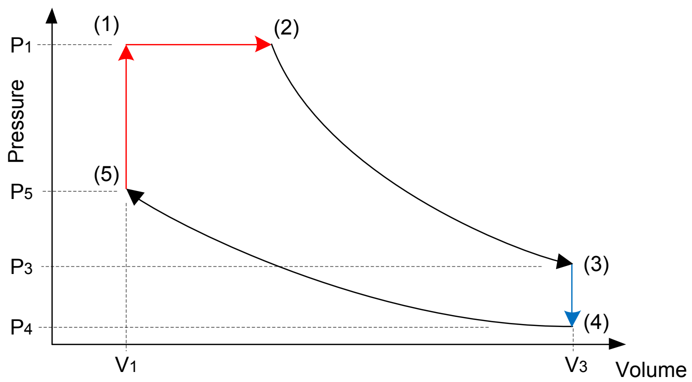

Figure 4. Pressure-volume indicated diagram with incomplete expansion and compression, considering inlet steam along the stroke of the piston (Badami and Mura, 2009).

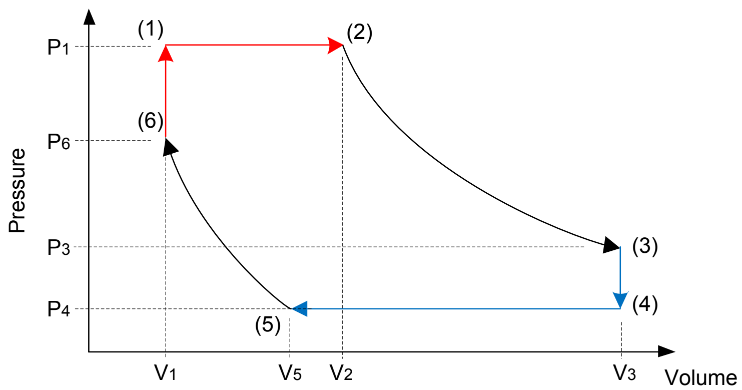

Figure 5. Generalized pressure-volume indicated diagram for an ideal cycle considering the incomplete expansion and compression (Antonelli and Martorano (2012) and Basbous et al. (2012)).

2.2. Equations and analysis

In order to analyze the proposed cycle of Fig. 5, it is necessary to define some relationship between the volumes of the cylinder, so it becomes easier to analyze and choose which volumes must be used to get better efficiency. Is defined as the cut-off ratio (r_C) the ratio between the volumes V_2 and V_1. The closure ratio (r_F) is defined as the ratio between V_5 and V_1. And the volume ratio (r_V) is defined as the relation between the maximum and the minimum volume, V_3 and V_1 respectively.

$$r_C = \frac{V_2}{V_1} \tag{1}$$

$$r_F = \frac{V_5}{V_1} \tag{2}$$

$$r_V = \frac{V_3}{V_1} \tag{3}$$

Furthermore, it is appropriate to express the cut-off ratio and the closure ratio in percentage form, where zero (0) means the dead volume and one (1) means the maximum volume within the cylinder. Therefore, these ratios can be written as:

$$r_{Cx} = \frac{r_C - 1}{r_V - 1} \tag{4}$$

$$r_{Fx} = \frac{r_F - 1}{r_V - 1} \tag{5}$$

It also becomes appropriate to define a pressure ratio, defined as the ratio between de working pressure of the evaporator and the condenser, given by Eq. (6):

$$r_P = \frac{P_{evap}}{P_{cond}} \tag{6}$$

The total work by cycle (W_{cycle}) can be calculated by the internal area of the diagram of Fig. 5. To define this equation, the steam was considered as a perfect gas with constant properties. The processes (1-2) and (4-5) are considered isobaric processes, the processes (2-3) and (5-6) are considered isentropic processes, and to calculate the work of the isentropic processes, this was considered a polytropic process, where the polytropic exponent is the adiabatic gas constant (k). The adiabatic gas constant is given by Eq. (7), where C_P and C_V are the specific heats at constant pressure and constant volume, respectively. Finally, the processes (3-4) and (6-1) are considered isochoric. There are then two isobaric processes, two isochoric processes and two isentropic processes. Since the total work is the sum of the work of each process of the steam engine, it can be defines by Eq. (8).

$$k = \frac{C_P}{C_V} \tag{7}$$

$$W_{cycle} = \frac{V_1 \times P_{evap}}{k-1}\left((k-1)(r_C - 1) + r_C\left[1 - \left(\frac{r_C}{r_V}\right)^{k-1}\right] + \frac{1}{r_P}\left[(k-1)(r_F - r_V) - r_F\left(r_F^{k-1} - 1\right)\right]\right) \tag{8}$$

One way of evaluating the efficiency of the cycle is through the specific work, but it is necessary to find the mass flow through the machine in a cycle. The mass spend in one cycle can be calculated as the difference between maximum and minimum mass inside the cylinder during one cycle. The mass contained in the cylinder at point (2) is the same at the point (3), which is the maximum mass in the cylinder (m_2). The mass contained in the cylinder at point (5) is the same at the point (6), which is the minimum mass in the cylinder (m_5). Thus the mass spent per cycle (m_{cycle}) can be written as:

$$m_{cycle} = m_2 - m_5 = \frac{V_1 \times P_{evap}}{k \times R \times T_{evap}}\left[k(r_c - 1) + 1 - \frac{{r_F}^k}{r_P}\right] \tag{9}$$

Where T_{evap} is the working temperature of the evaporator and R is the difference between the two specific heats, given by Eq. (10):

$$R = C_P - C_V \tag{10}$$

This way the specific work of the cycle (w_{cycle}) is:

$$w_{cycle} = \frac{W_{cycle}}{m_{cycle}} \tag{11}$$

The maximum possible work to be done by a machine is if it works isentropically, (i.e., causes an isentropic expansion on the steam). According to Sonntag et al. (2003), the isentropic specific work of an ideal gas is:

$$w_{iso} = -\frac{kR}{k-1}\left(T_f - T_i\right) \tag{12}$$

The isentropic specific work can be written as a function of the temperature of the evaporator and the volume ratio.

$$w_{iso} = \frac{kR}{k-1} T_{evap}\left(1 - \frac{1}{r_P^{\frac{k-1}{k}}}\right) \tag{13}$$

2.3. Efficiency

The isentropic efficiency (η_{iso}) of the machine is given by the relation between the specific work by the engine and the isentropic specific work of an ideal gas.

$$\eta_{iso} = \frac{w_{cycle}}{w_{iso}} \tag{14}$$

For this study, it is considered that at point (1) of all cycles, the pressure and the temperature within the cylinder are equal to the pressure and the temperature of the evaporator.

The calculations made through the above equations show that the isentropic efficiency of the cycle of Fig. 1 tends to one (1) when the dead volume tends to zero (0), and the larger the dead volume, lower the efficiency. The cycle shown in Fig. 2 has always an isentropic efficiency equal to 1, independently of the dead volume, and tends to the diagram of Fig. 1 when the dead volume tends to zero. The cycle of Fig. 3 has always an isentropic efficiency smaller then 1, and tends to 1 when the difference between the maximum and the minimum pressures tends to zero. The cycle shown in Fig. 4 always has an isentropic efficiency smaller then 1. The cycle of the Fig. 5, as shown in the diagram, always has a isentropic efficiency smaller then 1, but changing the cut-off ratio and the closure-ratio in a way this cycles becomes the cycle in Fig. 2, it reaches an isentropic efficiency equal to 1.

Thus, it can be noted that the efficiency of all cycles tends to 1, when the cycles tends to the cycle of the Fig. 2. it makes this diagram the ideal when considering the best efficiency. As mentioned, in practice is very difficult to make a machine that operates according to the cycle of Fig. 2, for reasons mentioned above.

Noting that in these diagrams, one point does not correspond to a thermodynamic state, since the mass within the cylinder varies. And for this reason it is not interesting study the indicated diagram using the specific volume, since not always a change in specific volume means that the piston is producing work.

If the generalized cycle shown in Fig. 5 is used in a design of a steam engine, one must be careful with the point of opening and closing the valves. If de cut-off ratio and the closure ratio are chosen inappropriately, there may be an over-compression (when the steam is compressed to pressures above the inlet pressure) or an over-expansion (when the steam is expanded to pressures below the condenser pressure) or cause it to work as compressor. A cycle with over-expansion and over-compression is shown in Fig. 6. If the steam is compressed to pressures above the intake pressure or expanded to pressures below the condenser pressure, the cycle efficiency will decrease due to the decreased area of useful work (7-2-8-5), besides the formation of areas where the work is negative (area 7-6-1) and (8-4-3). Depending on the opening and closing position of valves, the steam engine can even work as a compressor.

To prevent this to happening, it is necessary that the inlet valve closes after a certain position (closer to the BDC) and the exhaust valve closes before a certain position (closer to the TDC). These positions are called r_{Cmin} and r_{Fmax}, respectively, and may be written as:

$$r_{Cmin} = \frac{r_V}{r_P^{1/k}} \tag{15}$$

$$r_{Fmax} = r_P^{1/k} \tag{16}$$

Note that these values of r_C and r_F are such that, in the expansion process, the steam reaches the condenser pressure exactly at the BDC and in the compression process, the steam reaches the evaporator pressure exactly at the TDC.

Therefore, if the valves open and close at these points during a cycle, it becomes the cycle of Fig. 2, and it has the highest possible isentropic efficiency.

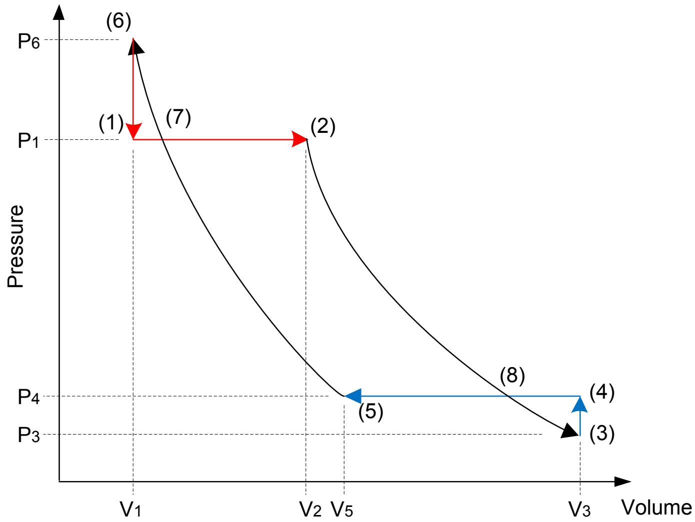

Figure 6. Pressure-volume indicated diagram with over-compression and over-expansion.

Using the software EES, although having an isentropic efficiency equal to one (1), the ideal cycle of Fig. 2 was studied to verify if it is still possible to optimize it. The calculations were made for an engine with an 8 cm diameter piston, the minimum distance between the piston and the cylinder head (dead space) of 3 cm, working with an absolute pressure in the condenser (P_{cond}) of 100 kPa and a working temperature of the evaporator (T_{evap}) of 350°C.

The results are shown in Fig. 7 and Fig. 8, noting that all point in these diagrams have the maximum possible isentropic efficiency. For both figures, it is noted that for a fixed volume ratio, there is a pressure that results in the maximum work done and the maximum mass consumption in one cycle. However, for a fixed pressure, the higher the volume ratio, higher the work done and the mass consumption in one cycle.

The operation pressure of the evaporator which results in the maximum work done is different from the operation pressure which results in the maximum mass consumption. The advantage of working at the pressure which results in maximum work per cycle is that the engine can operate at lower speeds or it is possible to achieve a higher output power at the same speed.

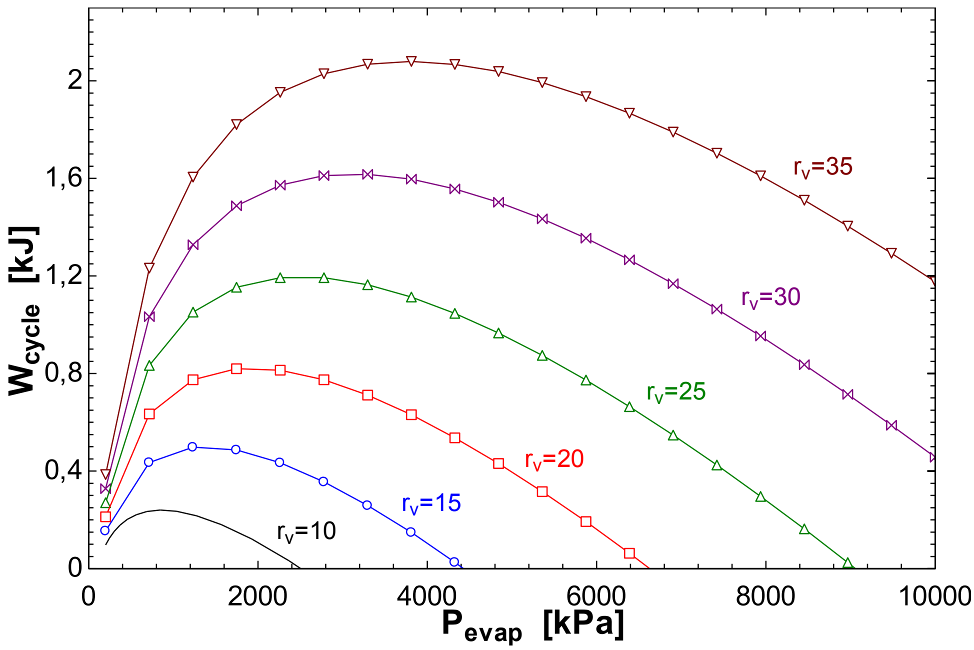

Figure 7. Work done in one cycle of operation for different volume ratios as a function of the pressure of the evaporator.

Considering that neither over-expansion nor over-compression occurs, the minimum isentropic efficiency occurs when the intake valve closes at the BDC and the exhaust valve closes at the TDC. In this case, the minimum isentropic efficiency can be written as:

$$\eta_{min} = \frac{(k-1)(r_V-1)\left(1-\frac{1}{r_P}\right)}{\left(k(r_V-1)+1-\frac{1}{r_P}\right)\left(1-\frac{1}{r_P^{\frac{k-1}{k}}}\right)} \tag{17}$$

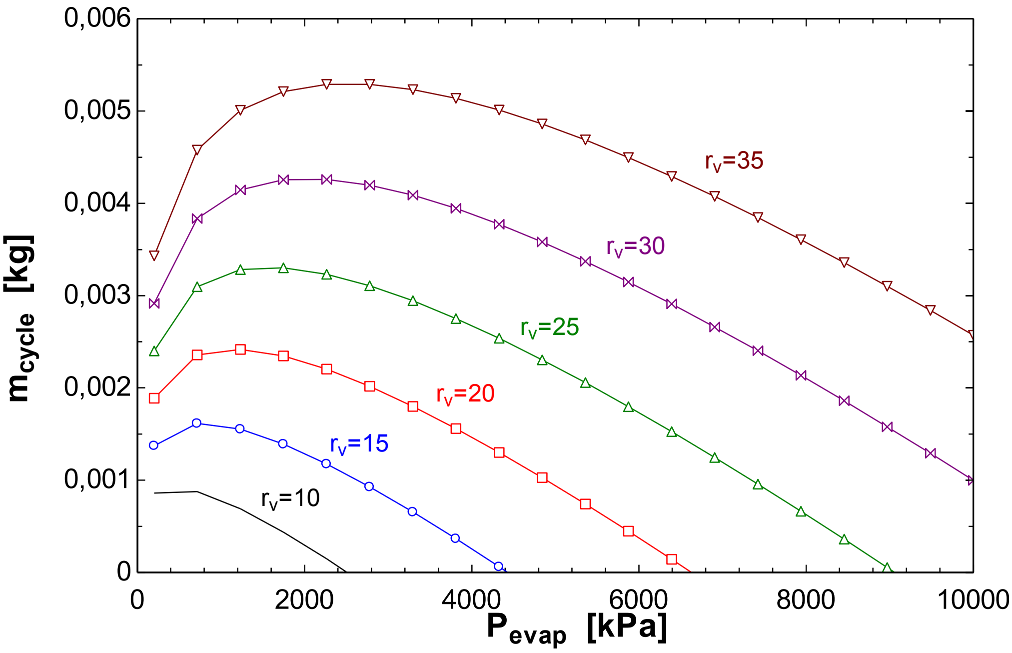

Figure 8. Mass consumption in one cycle of operation for different volume ratios as a function of the pressure of the evaporator.

2.4. Irreversibilities

In all cycles studied, all processes are adiabatic and the cycle showed in Fig. 2 is also reversible, which also means that all processes in Fig. 2 are isentropic within the cylinder, and hence this cycle has always an isentropic efficiency equal to 1. For the others cycles, which have a lower efficiency, will be studied the generalized cycle of Fig. 5.

The processes at constant volume are not isentropic, i. e. in those processes a quantity of energy that shall produce useful work is wasted. The process 6-1 is not isentropic because there is a mixture of gases at different pressures and temperatures within the cylinder. The process 3-4 is isentropic inside the cylinder, but when the gases exit the cylinder at a pressure greater than the condenser pressure, there is an increase in entropy due to the mixing of gases at different pressures and temperatures.

3. CONCLUSION

A generalized standard cycle for a reciprocating steam engine was defined (Fig. 5) and equated, since this cycle is the closest to the actual functioning of the engine and allows the study of the points of opening and closing the valves. However, the cycle with the maximum possible efficiency is the cycle of Fig. 2, and it is desirable that the steam engines or other expansion engines work in accordance with it, although this is very difficult in practice.

In the operation of expansion engines, such as steam engines, should be avoided isochoric processes, since they always cause useful energy waste.

4. ACKNOWLEDGEMENTS

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