OPTIMIZATION PERFORMANCE OF A MICROFLUID FLOW POWER CONVERTER

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Abstract

This article deals with an application of the endoreversible thermodynamics theory of heat engine applied to microfluid flow power converters (MFPC). An analogy is demonstrated between thermal and fluid flow efficiencies. Maximum power output and efficiency at maximum power are established for the device based upon the higher and lower pressure bounds. The linear and non linear fluid flows are considered with and without fluid friction losses. This paper provides theoretical limits for designing power flow converter. The best performances are obtained for linear fluid flow without flow losses.

Keywords: finite time thermodynamics, fluid flow, optimization performance

1 Introduction

During the past decades, micromachining technology has become available to fabricate micron mechanicals systems. Micromachines have a major interest in many disciplines like aerospace, engineering and biology Löfdahl (1999). In this article we limit our discussion to power converters from fluid flow by an analogy with the endoreversible thermodynamic theory of heat engine. The study of irreversible thermodynamic cycles has been undertaken by many researchers like Chambadal (1957), Novikov (1958), Curzon (1975) and Bejan (1988). The efficiency of some heat engines was studied at maximum power conditions on the basis of Carnot cycle models with or without irreversibilities of finite rate heat transfer and internal irreversibilities of the working fluid as described in Chen (1994), Blank (1996) and Kodal (2000). An engine is a device which converts potential difference of internal energy (temperature) into work and then into mechanical power. Similarly, we apply this physical principle to a pneumatic machine which converts a pressure drop into mechanic work. Many authors have exploited this idea and have derived the maximum power for a fluid flow with linear and non linear relations of pressure drop against flow rate (see Radchenco, 1994; Bejan, 1996; Chen et al, 1999).

A power flow converter is designed between two limits: a maximum power conversion and a maximum efficiency which corresponds to a reversible operation and zero power. Each of these bounds implies a specific relation of power conversion on the pressure of the two reservoirs between which the converter operates. In the present paper the nature of fluid flow (i.e. linear or non linear) and the losses by friction between the piston and the cylinder are taking into account. All the expressions will be derived algebraically and illustrated numerically. The aim of this paper is to find the optimal values of speed and pressures to provide maximum power output under optimal efficiency.

2 Theoretical Model

2.1 General Theory

We consider a double acting piston cylinder as shown in Fig. 1. This linear pneumatic actuator consists of only a few parts and is corrosion proof. The filament wound barrel is non-corrosive and has a surface for free travel and minimum friction. Besides, the piston moves with friction thanks to the pressure difference $P_1 - P_2$ maintained between its two faces.

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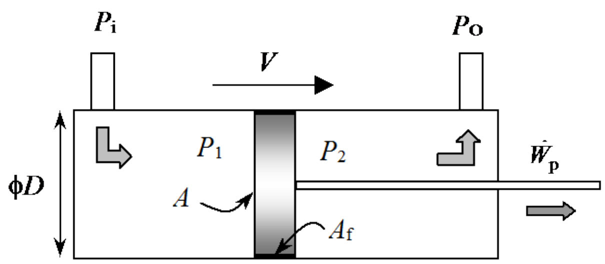

Fig. 1: *Microfluid flow power converter (MFPC)*

The working fluid (air) is admitted in the chamber with the pressure P_i while the left face of the piston is submitted to the pressure P_1. If we consider a non linear flow resistance relation (see Radchenco (1994) and Bejan (1996)), the pressure drop $(P_i - P_1)$ is written:

$$P_i - P_1 = K_1 V^n \quad (1)$$

Similarly, the fluid is ejected on right side of the piston with a pressure drop $(P_2 - P_o)$, where P_0 is the outlet pressure and:

$$P_2 - P_0 = K_2 V^n \quad (2)$$

V is the instantaneous linear speed of the piston, K_1 and K_2 are constant coefficients depending on duct and fluid properties and n is a coefficient which depends on the Reynolds number. The limit n = 1 corresponds to the linear flow when Eq. 1 and 2 obey the Poiseuille's law. The exponent n increases as the Reynolds number increases and for a non linear flow in the fully rough regime the value of the exponent is n = 2 as described in Idelchik (1993), Comolet (1994) and Faisandier et al. (1999).

The instantaneous power delivered by the piston to an external system is

$$\dot{W}_p = \left(P_1 - P_2\right) AV - \Delta P_f A_f V \quad (3)$$

where the product $\Delta P_f A_f V$ represents the fluid power loss by friction. ΔP_f is the friction pressure loss in the gap between the piston and the cylinder. A is the front area of the piston and A_f is the lateral friction area. If we consider the friction pressure is generated by a the viscous flow between two parallel surfaces ΔP_f is written:

$$\Delta P_f \approx \mu \frac{V}{d} \quad (4)$$

where μ is the coefficient of viscosity of the lubricant and d the relative motion gap thickness.

Let us consider the power $\dot{W}_i = P_i AV$ received from the fluid at the inlet of the actuator. Then, the energy conversion efficiency of the fluid flow power converter is:

$$\eta = \frac{\dot{W}_p}{\dot{W}_i} = \frac{\left(P_1 - P_2\right) AV - \Delta P_f A_f V}{P_i AV} = \frac{P_1 - P_2 - \dfrac{A_f}{A}\Delta P_f}{P_i} \quad (5)$$

The quantity $\dot{W}_i$ is analogous to the absorbed heat transfer rate of the heat engine (see Chen (1994), Kodal (2000)) and if we consider a reversible process without friction losses (μ = 0, then ΔP_f = 0) then $\dot{W}_p\big|_{rev} = \left(P_i - P_o\right) AV$ and the reversible efficiency of the fluid flow power converter is

$$\eta_{rev} = \frac{\dot{W}_p\big|_{rev}}{\dot{W}_i} = 1 - \frac{P_0}{P_i} \le 1 \quad (6)$$

We found an analogy between the Carnot efficiency of the heat engine and the reversible efficiency of the fluid flow power converter where the pressures are analogous to the temperatures (Fig. 2).

From Eq. 1 and 2 we obtain

$$P_1 = P_i - \frac{K_1}{K_2} P_2 + \frac{K_1}{K_2} P_0 \quad (7)$$

From Eq. 4 and 5,

$$P_1 = \eta P_i + P_2 + K_f V \ \text{ with } \ K_f = \frac{\mu}{d}\frac{A_f}{A} \quad (8)$$

Combining Eq. 7 and 8 gives

$$P_2 = \frac{P_i\left(1-\eta\right) + \dfrac{K_1}{K_2} P_0 - K_f V}{1 + \dfrac{K_1}{K_2}} \quad (9)$$

Substituting Eq. 9 into Eq. 7 yields

$$P_1 = \frac{P_i\left(\eta + \dfrac{K_2}{K_1}\right) + P_0 + K_f V}{1 + \dfrac{K_2}{K_1}} \quad (10)$$

Then, substituting Eq. 10 into Eq. 1 one obtains the piston speed by solving the general non linear equation:

$$\left(K_1 + K_2\right) V^n + K_f V + P_i\left(1-\eta\right) - P_0 = 0 \quad (11)$$

2.2 Linear Flow Resistance: n = 1

When the flow resistance is expressed by a linear relations (n = 1), the solving of the Eq. 11 becomes simple and the speed of the piston is

$$V_1 = \frac{P_i\left(1-\eta\right) - P_0}{K_1 + K_2 + K_f} \quad (12)$$

Combining Eq. 12 and 5 one finds the expression of the instantaneous power delivered by the piston to an external system

$$\dot{W}_{p1} = AP_i\eta\left[\frac{P_i\left(1-\eta\right) - P_0}{K_1 + K_2 + K_f}\right] \quad (13)$$

Equation 13 represents the instantaneous power versus the efficiency. The reservoir pressures P_i and P_0 are fixed. The power presents two limits:

$$\dot{W}_{p1} = 0 \quad (14a)$$

for

$$\eta = \eta_{min} = 0 \quad (14b)$$

and for

$$\eta = \eta_{max} = \eta_{rev} = 1 - \frac{P_0}{P_i} \qquad (14c)$$

Between these two limits, the piston output power has an optimal value. Taking the derivative of $\dot{W}_{p1}$ with respect to η and setting it to zero gives the maximum piston power output.

From Eq. 13, one solves $\frac{\partial \dot{W}_{p1}}{\partial \eta} = 0$ and finds the optimal value of the efficiency η_{opt1}

$$\eta_{opt1} = \frac{1}{2}\left(1 - \frac{P_0}{P_i}\right) \qquad (15)$$

It is easily demonstrated that the second derivative of $\dot{W}_p$ with respect to η is negative, $\frac{\partial^2 \dot{W}_{p1}}{\partial \eta^2} < 0$, so the optimal value of the efficiency η_{opt} corresponds to a maximum. Substituting Eq. 15 into Eq. 9, 10, 12, 13 gives the optimal values of pressures P_1 and P_2, piston speed V and power $\dot{W}_p$:

$$P_2\big|_{opt1} = \frac{\frac{1}{2}\left(P_i + P_0\right) + \frac{K_1}{K_2} P_0 - \frac{K_f}{2} \frac{\left(P_i - P_0\right)}{\left(K_1 + K_2 + K_f\right)}}{1 + \frac{K_1}{K_2}} \qquad (16)$$

$$P_1\big|_{opt1} = \frac{\frac{1}{2}\left(\frac{K_1 + K_2 + 2K_f}{K_1 + K_2 + K_f}\right)\left(P_i - P_o\right) + P_i \frac{K_2}{K_1} + P_0}{1 + \frac{K_2}{K_1}} \qquad (17)$$

$$V_{opt1} = \frac{P_i - P_0}{2\left(K_1 + K_2 + K_f\right)} \qquad (18)$$

$$\dot{W}_p\big|_{opt1} = \frac{A}{\left(K_1 + K_2 + K_f\right)}\left(\frac{P_i - P_0}{2}\right)^2 \qquad (19)$$

2.3 Non Linear Flow Resistance: n = 2

At higher Reynolds numbers the flow resistance is non linear. The solution of Eq. 11 with $n = 2$ gives the piston speed:

$$V_2 = \frac{-K_f + \sqrt{{K_f}^2 - 4\left(K_1 + K_2\right)\left(P_0 - P_i\left(1 - \eta\right)\right)}}{2\left(K_1 + K_2\right)} \qquad (20)$$

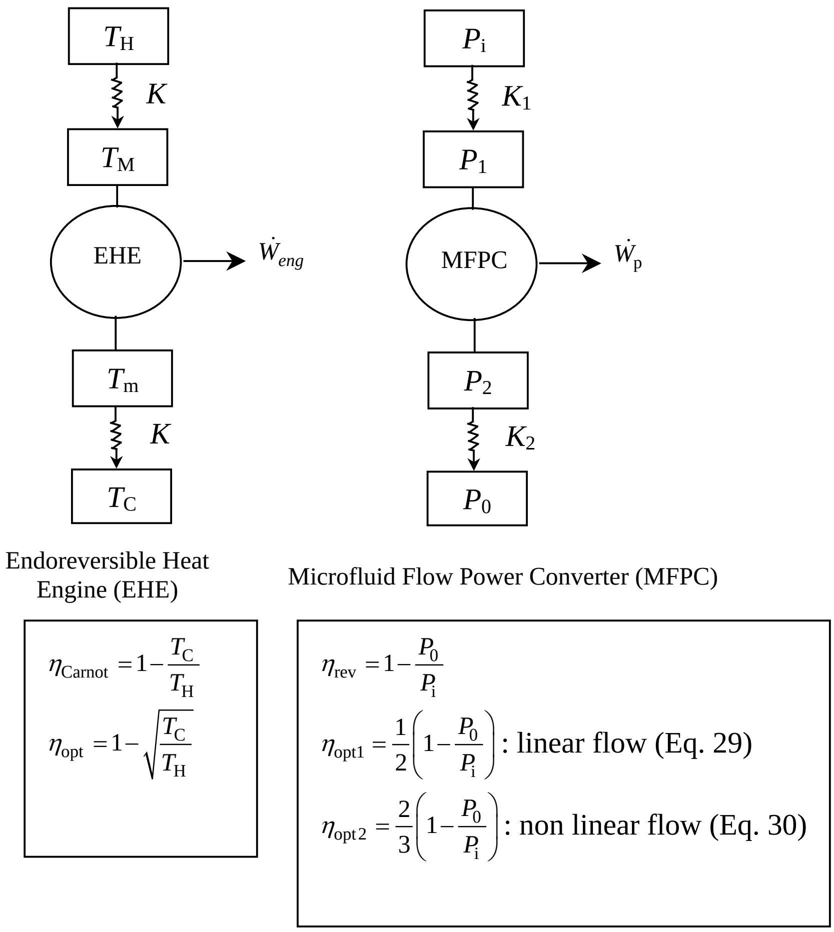

Fig. 2: *Analogy between an endoreversible heat engine (EHE) and a microfluid flow power converter (MFPC) without flow losses*

Combining Eq. 20 and 5, one obtains the expression of the instantaneous power delivered by the piston:

$$\dot{W}_{p2} = AP_i\eta\left[\frac{-K_f + \sqrt{K_f^{\,2} - 4(K_1+K_2)(P_0 - P_i(1-\eta))}}{2(K_1+K_2)}\right] \quad (21)$$

Equation 21 represents the instantaneous power versus the efficiency. The reservoir pressures P_i and P_0 are fixed. The power presents two limits:

$$\dot{W}_{p2} = 0 \quad (22a)$$

for

$$\eta = \eta_{min} = 0 \quad (22b)$$

and for

$$\eta = \eta_{max} = \eta_{rev} = 1 - \frac{P_0}{P_i} \quad (22c)$$

These two limits are not dependent on the friction losses K_f , besides, between η_{min} and η_{max} the power presents a maximal value for η_{opt2}:

$$\eta_{opt2} = \frac{K_f^2 - K_f\sqrt{K_f^2 + 3(K_1+K_2)(P_i - P_0)} + 6(K_1+K_2)(P_i - P_0)}{9(K_1+K_2)P_i} \quad (23)$$

Substituting Eq. 23 into Eq. 9, 10, 20 and 21 gives the optimal values of piston speed V, pressures P_1 and P_2, and power $\dot{W}_p$:

$$V_{opt2} = \frac{\sqrt{5K_f^{\,2} - 4K_f\sqrt{K_f^{\,2} + 3(K_1+K_2)(P_i - P_0)} + 12(K_1+K_2)(P_i - P_0)}}{6(K_1+K_2)} \quad (24)$$

$$P_1\Big|_{opt2} = \frac{P_i\left(\eta_{opt2} + \frac{K_2}{K_1}\right) + P_0 + K_f V_{opt2}}{1 + \frac{K_2}{K_1}} \quad (25)$$

$$P_2\Big|_{opt2} = \frac{P_i(1-\eta_{opt2}) + \frac{K_1}{K_2}P_0 - K_f V_{opt2}}{1 + \frac{K_1}{K_2}} \quad (26)$$

$$\dot{W}_p\Big|_{opt2} = AP_i\eta_{opt2}\left[\frac{\sqrt{K_f^{\,2} + 4(K_1+K_2)(P_0 + P_i(1-\eta_{opt2}))} - K_f}{2(K_1+K_2)}\right] \quad (27)$$

3 Discussion

In order to show the applicability of the model we analyzed the two limit cases, the linear flow (n = 1) and the non linear flow (n = 2) from a parametric study. Although many MEMS devices use different working fluids, dry air is used in this work. The outlet reservoir pressure P_0 is set to 1 bar. The flow coefficients K_1 and K_2 are chosen to be 4x10^4 to match typical flows and geometrical characteristics encountered in MEMS technology. The coefficient of friction K_f is in the range 0 to infinity where the values 0 and infinity correspond to the cases of none and maximal friction losses respectively. Between these two values we set K_f to 4x10^4 to conserve the same influence on the pneumatic characteristics of the device than the flow coefficients K_1 and K_2.

All expressions concerning the piston speed, the gas pressures, the power and the efficiency, depend on the fluid and geometrical flow resistances K_1, K_2 and K_f except for the linear flow optimal efficiency η_{opt1}. These expressions are compared with those obtained by the two different investigators Bejan (1996) and Chen et al (1999).

3.1 Optimal Values

The optimal values of pressure, speed, power and efficiency are plotted versus reservoir pressure P_1. Results are presented on different diagrams. Optimal values present an increase with respect to the input pressure and we observe the bad influence of the irreversible friction losses K_f on the general performances of the converter (Fig. 3 to 7). If $K_f \to \infty$: these optimal values tend to zero, friction losses being very important, except for the optimal efficiency.

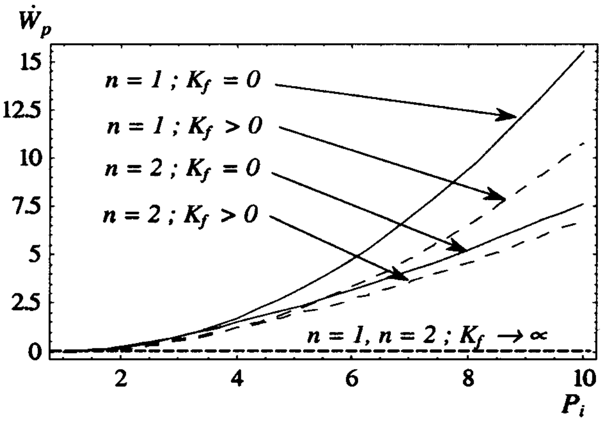

Fig. 3: *Optimal power versus input pressure. Linear flow: n = 1, non linear flow: n = 2, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

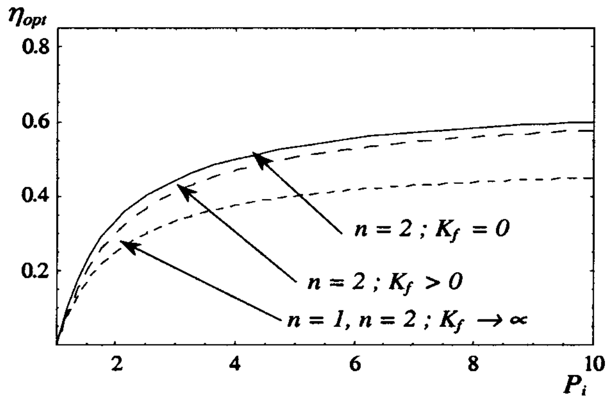

Fig. 4: *Optimal efficiency versus input pressure. Linear flow: n = 1, non linear flow: n = 2, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

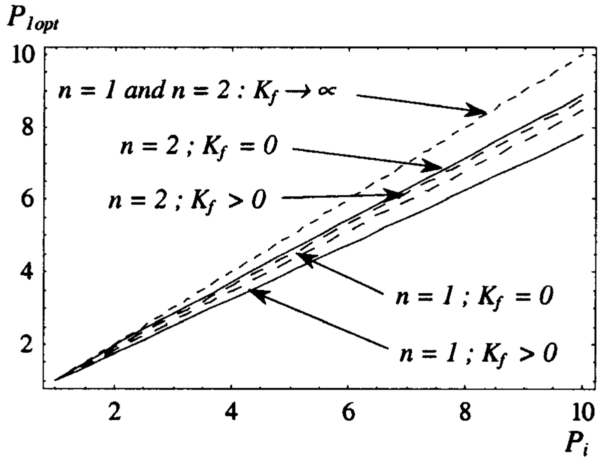

Fig. 5: *Optimal pressure P_1 versus input pressure. Linear flow: $n = 1$, non linear flow: $n = 2$, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

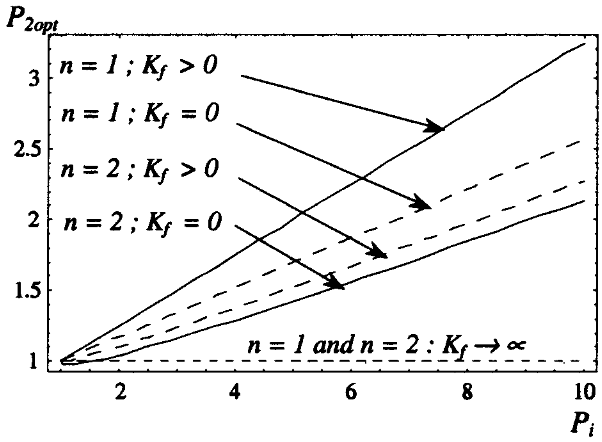

Fig. 6: *Optimal pressure P_2 versus input pressure. Linear flow: $n = 1$, non linear flow: $n = 2$, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

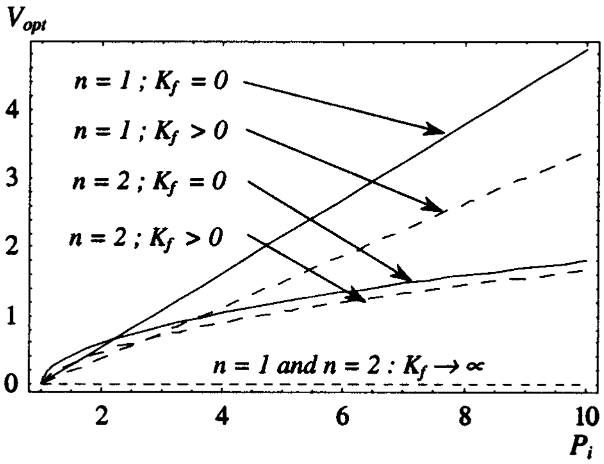

Fig. 7: *Optimal piston velocity versus input pressure. Linear flow: $n = 1$, non linear flow: $n = 2$, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

Sample results are shown on Fig. 4 where the efficiency is plotted versus input reservoir pressure for several values of losses characteristics. The efficiency increases with the input reservoir pressure. One demonstrated that only the optimal efficiency η_{opt1} (Eq. 15) is independent of the flow characteristics K_1 and K_2 when the flow is linear and one found expressions consistent with or complementary to those obtained by Bejan (1996) and Chen et al (1999).

If one considers friction losses are negligible (K_f = 0), Bejan (1996) and Chen et al (1999) demonstrated that for a linear flow (n = 1) and for a non linear flow (n = 2).

$$\eta_{\text{opt}} = \frac{n}{n+1}\left(1-\frac{P_0}{P_i}\right) \tag{28}$$

In this work, the maximal efficiency becomes similar to Eq. 28 and from Eq. 15 and 23 with $K_f = 0$ for a linear flow, n = 1 in Eq. 28,

$$\eta_{\text{opt1}} = \frac{1}{2}\left(1-\frac{P_0}{P_i}\right) \tag{29}$$

In this case, the optimal power efficiency is exactly the half of the reversible efficiency η_{rev} (Eq. 6) for a non linear flow, n = 2 in Eq. 28,

$$\eta_{\text{opt2}} = \frac{2}{3}\left(1-\frac{P_0}{P_i}\right) \tag{30}$$

Equation 29 and 30 represent the quality of the thermomechanical power conversion in the optimal flow conditions. These two expressions are analogous to the maximal efficiency of a heat engine without internal irreversibilities founded by Curzon (1975) and Kodal (2000).

$$\eta_{\max} = 1-\sqrt{\frac{T_C}{T_H}} \tag{31}$$

where T_C and T_H are the cold sink and the hot source temperatures respectively.

Optimal pressures and piston speed expressions (Eq. 16 to 18 and Eq. 24 to 26) are consistent to those obtained by Chen (1999) in both cases of linear and non linear flows. The efficiency decreases if friction losses are not negligible, (K_f > 0) but if $K_f \to \infty$, the corresponding curve presents an evolution identical to the linear case described by Eq. 29. In this case, the piston moves slowly and therefore the flow becomes linear. K_f is analogous to the internal irreversibility of the heat engine (see Chen et al (2001), Gordon (1992) and Radcenco et al (1993)).

3.2 Output Power and Speed versus Efficiency

The qualitative behavior presented in Fig. 8 appears to be common to almost thermal converters like heat engines (see Curzon (1975), Blanck (1996), Chen (1994), Kodal (2000), Chen et al (2001), Gordon (1992) and Radcenco (1993)), thermoelectric generators (Nuwayhida et al (2000), Chen (2000)) chemical processes (Chen (1998) and Le Goff (2000)). The loop behavior in Fig. 8 can be understood from analyzing the two limits of low and high pressure reservoir operation. At low pressures or when $P_i = P_1$ both power and efficiency vanish because the piston does not move: there is no pressure difference between its two faces. At this point, the pressure is shunted directly from the high pressure level reservoir to the low pressure level reservoir with no net power produced. In the theory of heat engine, this point corresponds to the called thermal short-circuit limit described by Gordon (1992). At high pressure, fluid frictions increases more rapidly than the power output, so power output vanishes at finite rates of pressure input. Hence efficiency vanishes too because the reversible limit is reached with maximal efficiency (Eq. 14 and 22). A reversible cycle takes infinite time or infinite heat exchange surface area to complete and consequently the output power falls down. Fig. 8 shows that an increase in the friction coefficient K_f results in an decrease of the output power $\dot{W}_p$ both for the linear and the non linear flows. The power converter is more efficient if the flow is linear because the power output is greater than in the non linear flow case due to the increase of fluid friction. If $K_f \to \infty$ the power output tends to zero, friction losses being very important. In MEMs technology, the dimensions are very small then the frictions losses become very important. Figure 8 and 9 show that the increase in the friction coefficient K_f creates power loss and speed loss more important in the case of linear flow than in the case of non linear flow.

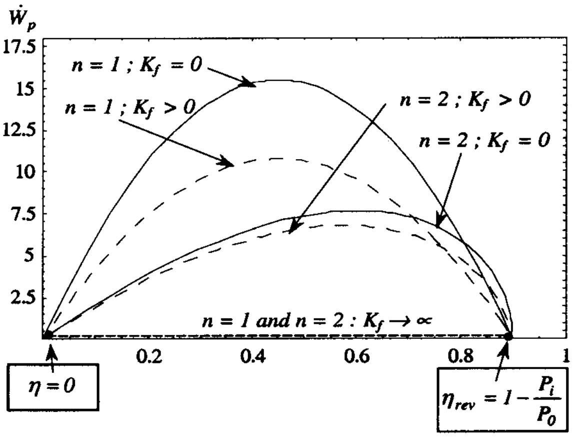

Fig. 8: *Power versus efficiency. Linear flow: n = 1, non linear flow: n = 2, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

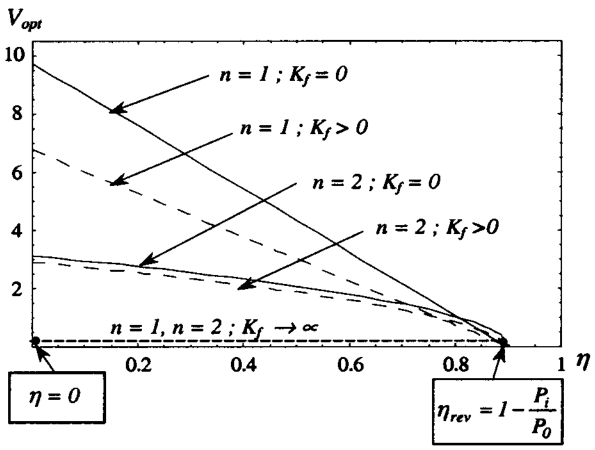

Fig. 9: *Optimal speed of the piston versus efficiency. Linear flow: n = 1, non linear flow: n = 2, $K_1 = K_2 = 4x10^4$ and $K_f = 4x10^4$ for the case $K_f > 0$*

4 Conclusion

This paper analyzed the accuracy of the optimization of a power fluid converter by considering the results obtained between two limits: the linear and the non linear flows with and without flow losses. We showed the analogy between the fluid power converter and the heat engine by the way of a theoretical development in finite time thermodynamics analogy. Flow losses affect the performances of the device and must be considered in the analysis. In each case of flow the optimal values of speed, power, pressures and efficiency are established. We demonstrated the bad influence of the friction losses, especially in MEMs technology, both for the linear and the non linear flows on the maximal power output under optimal efficiency. This paper provides theoretical limits for designing power flow converter.

Nomenclature

A	Front area of the piston	[m^2]
A_f	Lateral friction area	[m²]
D	Relative motion gap thickness	[m]
D	Diameter of the cylinder	[m]
K_f	Fluid conductance	[Pas/m]
K_m	Thermal conductance at the cold sink	[W/K]
K_M	Thermal conductance at the hot source	[W/K]
K_1	Fluid conductance at the inlet of the actuator	[-]
K_2	Fluid conductance at the outlet of the actuator	[-]
N	Coefficient depending of the nature of the flow	[-]
P_i	Inlet pressure of the reservoir	[Pa]
P_o	Outlet pressure of the reservoir	[Pa]
P_1	Pressure at the left face of the piston	[Pa]
P_2	Pressure at the right face of the piston	[Pa]
$P_1\|_{opt1}$	Optimal pressure at the left face of the piston for linear flow	[Pa]
$P_2\|_{opt1}$	Optimal pressure at the right face of the piston for linear flow	[Pa]
$P_1\|_{opt2}$	Optimal pressure at the left face of the piston for non linear flow	[Pa]
$P_2\|_{opt2}$	Optimal pressure at the right face of the piston for non linear flow	[Pa]
T_C	Temperature of the cold sink	[K]
T_H	Temperature of the hot source	[K]
V	Speed of the piston	[m/s]
V_{opt1}	Optimal speed of the piston for linear fluid flow	[m/s]
V_{opt2}	Optimal speed of the piston for non linear fluid flow	[m/s]
V_1	Speed of the piston for linear fluid flow	[m/s]
V_2	Speed of the piston for non linear fluid flow	[m/s]

$\dot{W}_{\mathrm{i}}$	Fluid power at the inlet of the actuator	[W]
$\dot{W}_{\mathrm{p}}$	Instantaneous power delivered by the piston	[W]
$\dot{W}_{\mathrm{p}}\big\vert_{\mathrm{opt1}}$	Optimal piston power for linear fluid flow	[W]
$\dot{W}_{\mathrm{p}}\big\vert_{\mathrm{opt2}}$	Optimal piston power for non linear fluid flow	[W]
$\dot{W}_{\mathrm{p}}\big\vert_{\mathrm{rev}}$	Reversible piston power	[W]
ΔP	Pressure difference	[Pa]
ΔP_{f}	Pressure difference losses due to piston-cylinder friction	[Pa]
η	Efficiency	[-]
η_{Carnot}	Carnot efficiency	[-]
η_{min}	Efficiency	[-]
η_{max}	Efficiency at maximum power	[-]
η_{opt}	Optimal efficiency	[-]
η_{opt1}	Optimal efficiency for linear fluid flow	[-]
η_{opt2}	Optimal efficiency for non linear fluid flow	[-]
η_{rev}	Efficiency for reversible conditions	[-]
μ	Dynamic viscosity of the lubricant	[Pas]

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