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Analysis of Vortex Tubes combined with an Energy Regeneration System in Vehicular Air Conditioning

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Abstract. *Vehicular air conditioning systems have always utilized the thermodynamic vapor-compression cycle concept. Then, the investigation of other refrigeration mechanisms are welcome. One of these possibilities is the application of the vortex tubes, that work with the introduction of an air compressed flow. Once there is not the necessity of heat exchangers and refrigerant fluids, this device enables a higher flexibility in the positioning of air outlets in some cabin position. The feeding of vortex tubes occurs only with compressed air flow, which can be stored for consumption when necessary. In addition, because of this characteristic, there is the possibility of the energy recovery, that is dissipated to the environment in the vehicle braking process currently. This study has been divided in the investigation of vortex tubes and regenerative systems, investigation of the potential of energetic reuse, configuration system proposal, main components first selection and viability analysis for cars and trucks. The viability and performance analysis was made based on a typical operation cycle for typical cars and trucks. The results showed the dificulties in the application of this philosophy for light vehicles with more viability for commercial vehicles use.*

Keywords*. Vortex Tubes, Air Conditioning, Motor Vehicles, Energetic Regeneration, Braking.*

1. Introduction

According to Kaynakli and Horuz (2003) the vehicular air conditioning system, before treated as standard feature only in high-series vehicles, has become a standard feature not only in light passenger vehicles but also cargo vehicles. Not only is the comfort optimized but also the security of the occupant. This can be explained by the fact that the glasses are kept closed, minimizing the action of thieves and the thermal stress of the driver, increasing his alert level in the traffic.

A conventional vehicular air conditioning system operates as a refrigeration cycle and therefore the supply of work to the system is needed. Conventionally, such work is injected in the system by the compressor. This work is obtained through the kinetic energy transference of internal combustion engine of the vehicle.

In such a way, the work executed by the compressor reflects in an increase of fuel consumption, affecting the global efficiency of the internal combustion engine of the vehicle. Beyond the fuel consumption increase, there is also an available power reduction that could be used for the vehicle movement.

Then, the study of alternative climatization devices in order to find a device that allies qualities as lower energy consumption and application flexibility is welcome. An alternative device that is studied in this paper is the vortex tube.

Cockerill (1998) mentions that the vortex tubes (also called Ranque-Hilsch Tubes) are simple devices that separate a primary gas flow in two secondary flows: an outlet high temperature flow (higher temperature than the inlet flow) and an outlet low temperature flow (lower temperature than the inlet flow). Ahlborn et al. (1994) exemplifies the magnitude of the temperature differential: for fluid condition of inlet pressure of 0.3 MPa and Temperature of 290K, is possible to extract flows with temperatures around 330K (high temperature flow) and 260K (low temperature flow). The simplicity of such devices is cited by Saidi and Valipour (2003), as presented in the Fig. (1).

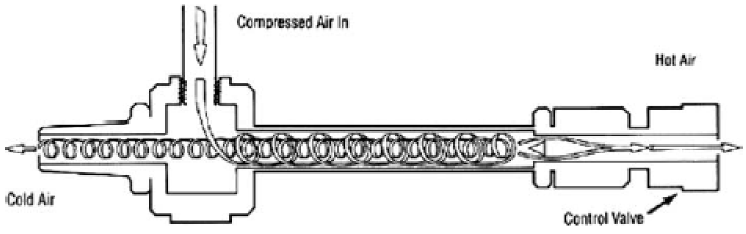


Figure 1. Example of vortex tube (Saidi and Valipour, 2003)

The energy consumption optimization for climatization applications can be executed by means of studying forms of energy reuse. Analyzing a conventional automobile, it is verified that energy is wasted at the moment of the vehicle braking. This energy could be reused to supply compressed air for the vortex tube operation. A possibility that will be studied in this paper is the utilization of a regenerative braking system that stores energy to be used when necessary.

2. The vortex tube

The vortex tube, also called Ranque-Hilsch Tube, is a thermomechanical device without mobile parts that allows the division of a main fluid flow into two secondary flows of distinct temperatures (Cockerill, 1998). Physically, such device consists of a cylindrical pipe with an inlet flow, located radially, so that the trajectory of flow has a tangent with the internal face of the pipe. The secondary fluid flows leave the vortex tube in the axial directions, that can be coincident or opposite directions, depending on the construction type, as cited by Cockerill (1998). Observing the Fig. (1), it is evident the non-existence of mobile parts in the vortex tube, minimizing the complexity. This characteristic composes one of the advantages of such devices. Another advantage constitutes the magnitude of the temperature differential of the outlet flows.

The work of Ranque (1933) was a pioneer study about vortex tubes. It mentions that the temperature differential depends on the relation between the outlet mass flows (cold flow and hot flow). Using a vortex tube with inlet tube of diameter 12mm and inlet pressure of 589 kPa, Ranque (1933) got a temperature differential of 70K, for the condition of minimum hot flow.

Later, Hilsch (1946) executed a detailed quantitative study in Vortex Tubes. He executed experiments changing many variables: diameter of the central orifice, inlet pressure and cold mass flow ratio (relation between the outlet cold flow rate and inlet flow rate). The Fig. (2) presents the vortex tube used by Hilsch (1946).

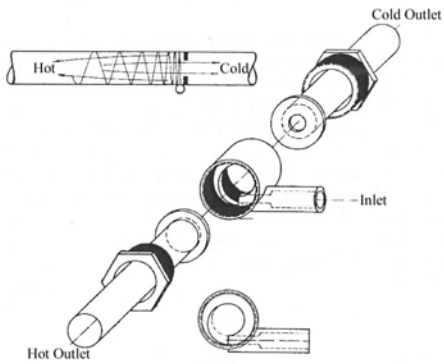


Figure 2. Vortex tube used by Hilsch (Hilsch, 1946)

After the study of Hilsch (1946) other studies have followed. Pengelley (1957) studied physical mechanisms involved in the phenomenon of energy separation. Sibulkin (1962) developed a theory in order to predict the characteristics of performance of the vortex tube. Soni and Thomson (1975) applied a methodology in order to identify the variables that present great influence in the performance of the vortex tubes. Takahama and Yokosawa (1981) studied the possibility to minimize the length of the chamber of the vortex tube without minimize the performance. Kurosaka (1982) presented a new theory that explains the energy separation: the acoustic waves would produce disordered disturbs in the air flow inducing the energy separation. Later, Stephan et al. (1983) developed a study of the energy separation process: the results showed the distribution of the temperature variation of the low-temperature air flow throughout the length of the chamber of the vortex tube. Stephan et al. (1984) established a similarity relation of the temperature variation of the cold flow as a function of the cold flow ratio in geometrically similar vortex tubes, using a mathematical formularization. The mathematical model was applied to diverse gases (air, Helium and Oxygen), and it was obtained the variation of the cold flow temperature as a function of the cold air flow ratio. Alhborn et al (1994) studied more recently the influence of the kinetic energy in the outlet hot and cold air flows. Alhborn, Camire and Keller (1996) analyzed the effect of the vortex tubes operation in low pressures. The search for new forms to optimize the vortex tubes efficiency led Piralishvili and Polyaev (1996) to study optimized pipe geometries of double-circuit vortex tubes. Saidi and Yazdi (1999) studied optimized dimensions and operation conditions, applying the exergy analysis. In order to evaluate the effect of the geometric parameter variation in the efficiency and the thermophysical parameters of the vortex tube, Saidi and Volipour (2003) developed an experimental model of a vortex tube for cooling, using variations of vortex tubes.

3. Braking regenerative systems

The constant changes in the environmental legislation in order to minimize the gases emissions have caused changes in the transport ways. Schaible and Szabados (1994) cite that a possible way to eliminate the gases emissions in the current engines would be the substitution of the internal combustion engines by electric engines. Another way to minimize the gases emissions is the use of regenerative braking systems.

In the braking moment of the vehicle, two thermodynamic processes are observed, as cited by Wicks and Donnelly (1997). The first thermodynamic process consists in the friction, in which the ordered energy of the vehicle is converted into disordered thermal energy in the brakes. This results in the temperature increasing of the brakes during the braking process. The second thermodynamic process consists in the heat transfer from the brakes to the environment. Then, as observed by Wicks and Donnelly (1997), there are irreversible exergy from the vehicle to the environment during the braking process. The regenerative braking systems work in the reutilization concept of part of the energy that would be wasted to the environment in the moment of the vehicle braking. Then, such energy can be reutilized later, minimizing the fuel consumption. Normally, the regenerative braking systems are applied in electric vehicles. These have already made use of an electric engine that can work in the opposite direction as a generator, as cited by Schaible and Szabados (1994). In this process, a torque is supplied to the generator, converting mechanical energy (deriving from the axle movement) into electric energy that is stored in batteries.

There are cases where the regenerative braking system is applied in conventional vehicles endowed of internal combustion engines. Chicurel (1999) cites an example of the use of a combined regenerative-dissipative braking system with a hydro-pneumatic accumulator. In this case the hydro-pneumatic accumulator executes the function of the batteries, storing energy in the form of pressurized fluid, to be used later.

4. Potential of energy regeneration in the vehicle braking

For the study of the available energy for regeneration, it was adopted two cycles of conduction: a cycle specified according the standard ABNT NBR 6601 with duration of 1370s (for passenger light vehicles) and a linear unacceleration from 40km/h – 0km/h @ 5s (passengers and commercial vehicles). The basic data of the vehicles, needed for the available energy simulation, is presented in the Tab. (1) (Itao, 2005).

Table 1 – Basic data for the simulation of the energetic regeneration potential

Vehicle Class	Vehicle Model	Mass m (kg)	Aerodynamic Drag Coefficient C_d (-)	Vehicle frontal projected Area A_f (m^2)	Tire pressure p_{tires} (psi)
Passenger Vehicles	Chevrolet Celta 1.0L 2P	834			
	Fiat Mille Fire Flex 1.0L 2P	810			
	Ford Ka 1.0L 2P	910			
	Volkswagen Gol Total Flex 1.0L 2P	861			
	Value adopted**	**854**	**0.34***	**~ 1.7**	**~ 28**
Commercial Vehicles	Scania R360 4x2	7227		~ 7,4	
	Volkswagen 18310 Titan 4x2	6163		~ 6,3	
	Volvo FH12 4x2	7100		~ 7,4	
	Value adopted**	**6830**	**0.90***	**~ 7.0**	**> 50**
Commercial Vehicle with a 15 ton. Trailer	**Value adopted**	**21830**	**0.70***	**~ 7.0**	**> 50**

* C_d estimated data according Götz e Mayr (1998).

** Weight value data was adopted based in the average value of the analysed vehicles.

For the energetic regeneration calculation, the energy balance was applied to the vehicle, as showed in the Eq. (1).

$$\tfrac{1}{2}.m.\left(v_2^2 - v_1^2\right) + m.g.\left(Z_2 - Z_1\right) + \Delta U_{vehicle} = Q - \left[\int_{v1}^{v2}\int_{s1}^{s2}\left(R_R + R_a + R_\alpha + R_E\right).dv.ds\right] \tag{1}$$

Where m is the mass (kg), v_2 and v_1 are respectively the final and initial speed (m/s), g is the standard acceleration of gravity (9.81 m/s^2), Z_2 and Z_1 are respectively the final and initial elevation (m), $\Delta U_{vehicle}$ is the vehicle internal energy variation (J), Q is the heat (J), R_R is the rolling drag (N), R_a is the aerodynamic drag (N), R_α is the ground inclination drag (N), R_E is the gearing drag (N) and s is the space (m).

Some assumptions were adopted in this study:

- There is no potential energy variation ($Z_2 – Z_1 = 0$) and the vehicle is running on a plane ($R_\alpha = 0$);
- There is no vehicle internal energy variation ($\Delta U = 0$);
- During the braking process the vehicle is out of gear ($R_E = 0$).

Applying the previous assumptions to Eq. (1), it results in Eq. (2), where E_{liq} is the net energy available for regeneration (J):

$$E_{liq} = Q = \tfrac{1}{2}.m.\left(v_2^2 - v_1^2\right) - \left[\int_{v1}^{v2}\int_{s1}^{s2}\left(R_R + R_a\right).dv.ds\right] \tag{2}$$

With R_a and R_R can be obtained applying the Eq. (3), Eq. (4) and Eq. (5), presented by Gillespie (1992):

$$R_a = \tfrac{1}{2}.\rho.v^2.C_d.A_f \quad (3)$$

$$R_R = f_r.m.g \quad (4)$$

$$f_r = f_0 + 3.24.f_s.(v/100)^{2.5} \quad (5)$$

Where ρ is the air density (kg/m^3), C_D is the aerodynamic drag coefficient (-), A_f is the vehicle frontal projected area (m^2), f_r is the global rolling coefficient (-), f_0 is the speed rolling coefficient (-) and f_s is the basic rolling coefficient (-).

The coefficients f_r and f_0 can be found according to Fig. (3), presented by Gillespie (1992).

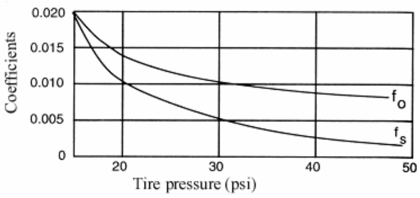


Figure 3. Rolling coefficients (Gillespie, 1992)

Solving the Eq. (2) with the conduction cycles data, it was obtained the results presented in the Fig. (4) and Fig. (5).

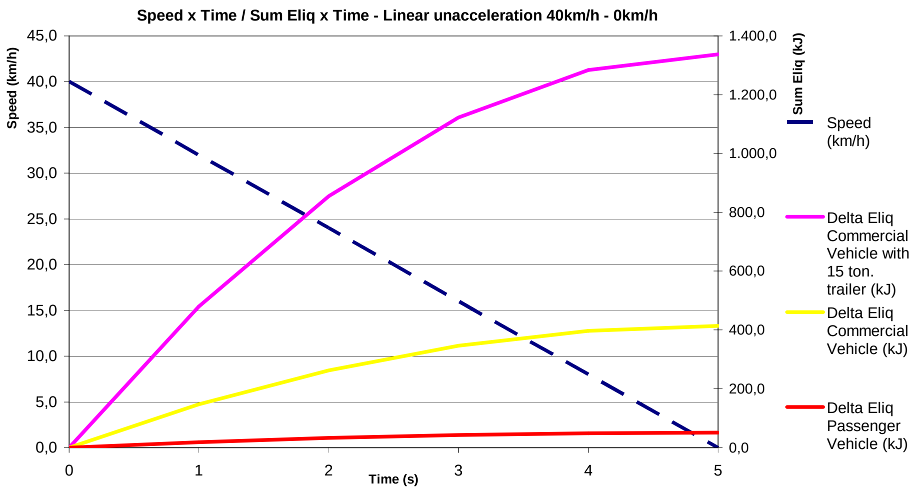


Figure 4. Speed variation and evolution of the available accumulated liquid energy as a function of the time (linear unacceleration cycle)

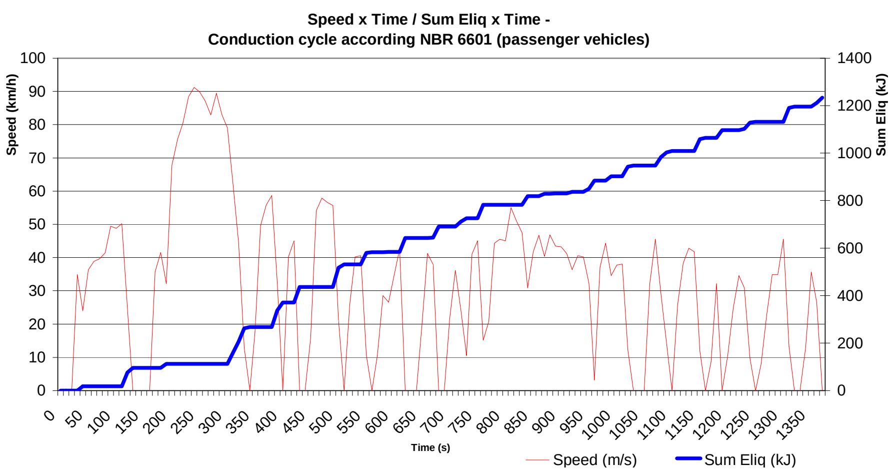


Figure 5. Speed variation and evolution of the available accumulated liquid energy as a function of the time (NBR 6601 cycle)

5. Study of an air conditioning system based on the application of vortex tubes and energetic regeneration

5.1. Operation philosophy

According to Cockerill (1998), the operation of a vortex tube becomes only possible, if the system has some basic components: air compressor, vortex tube and pipes.

In automobiles, the energy necessary for the operation of the compressor can be obtained through diverse sources, such as electric energy obtained through the vehicle alternator or kinetic energy obtained from the engine of internal combustion. A pneumatic accumulator can be added to the system, with the objective to supply an air reserve, making possible the system operation when the compressor is not operating.

In a conventional automotive air conditioning system, the cooling fluid compressor is driven by the internal combustion engine. Such philosophy naturally causes a fuel consumption increasing: the engine must burn fuel to vehicle’s motion and to the operation of the conventional air conditioning system.

However, as presented previously, there is a potential reuse from the kinetic energy that is wasted to the environment in the vehicle braking process. An optimized compressor operation hypothesis consists in the reuse of such kinetic energy, minimizing the energy consumption of the internal combustion engine.

To allow the maximum energy recovering, it’s recommendable to apply some energy accumulation device. Taking into account that the vortex tube operates only with the application of compressed air, a proposal would be the air compressor operation through the kinetic energy that would be rejected as heat during the vehicle braking process. In this case, the application of a device that transfers the vehicle kinetic energy, from the transmission to the compressor (only in the vehicle braking action), becomes necessary.

In this way, the system would be composed of the following components: air compressor, pipes, pneumatic accumulator for air storage, vortex tube that supplies hot air and cold air, internal combustion engine, engine transmission, clutch of the regenerative system (transmission), clutch of the regenerative system (engine), belts and tires.

When the air conditioning system is operating without energetic regeneration, the vortex tube will consume the compressed air of the pneumatic acumulator. Naturally, the consumption of air will lead to the emptying of the pneumatic acumulator, causing a pressure drop of the same. In this case, the compressor must receive kinetic energy from the vehicle engine, inevitably resulting in an additional fuel consumption.

Then, it is possible to identify three system operation modes. In order to simplify the understanding, Fig. (6) presents a flowchart with such operation modes

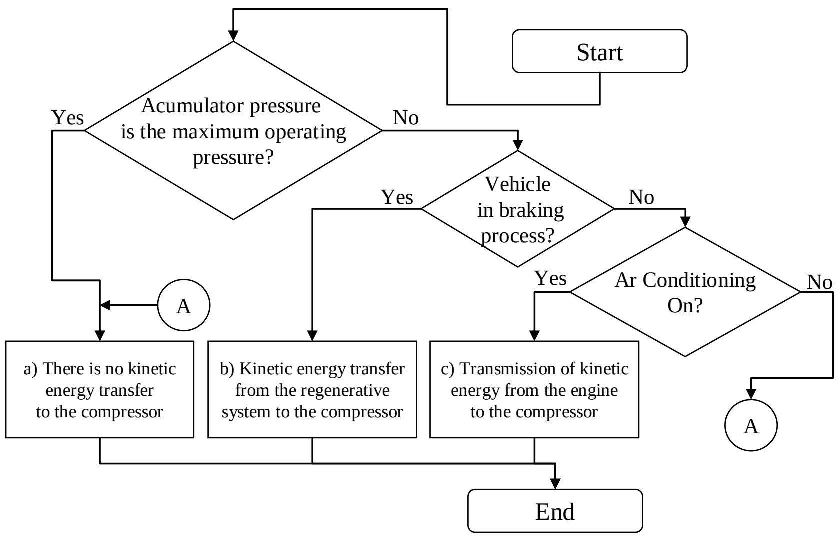
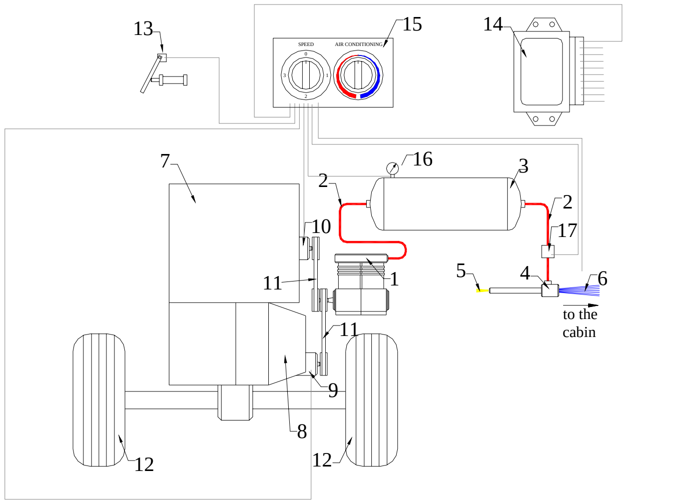


Figure 6. Operation modes flowchart of the air conditioning system with application of vortex tubes and energy regeneration

In order to assist the system operation in these modes, it is necessary the addition of sensors and modules. The new system becomes composed as described in the Fig. (7): air compressor (1), pipes (2), pneumatic accumulator (3), vortex tube (4) that supplies hot air (5) and cold air (6), internal combustion engine (7), engine transmission (8), clutch of the regenerative system (transmission) (9), clutch of the regenerative system (engine) (10), belts (11), tires (12), brake pedal module (13), engine ECU (14), air conditioning ECU (15), pressure sensor (16) and valve (17).



1. Compressor
2. Pipes
3. Pneumatic acumulator
4. Vortex tube
5. Hot air
6. Cold air
7. Internal combustion engine
8. Engine transmission
9. Regenerative system clutch - transmission
10. Regenerative system clutch - engine
11. Belt
12. Tires
13. Brake pedal module
14. Engine ECU
15. Air Conditing ECU
16. Pressure sensor
17. Valve

Figure 7. Proposal of air conditioning system with the application of vortex tubes and energy regeneration (with sensors)

In the sequence, it will be analised the function of the system components for each operation mode, described in Fig. (7):

a) No energy transfer to the compressor: this operation mode occurs basically when the pneumatic accumulator is in the full capacity (maximum operation pressure). In this situation, in the braking moment, there will not be the functioning of the regenerative system, that would transfer kinetic energy of the transmission of the vehicle to the compressor. This is possible because a pressure sensor (16) sends a signal to the air conditioning ECU, that sends a signal to clutches (9) and (10). The clutches do not allow the kinetic energy transfer through the belt movement (11) to the compressor (1).

b) Energy transfer from the regenerative system to the compressor: this operation mode occurs during the vehicle braking process, since the pneumatic accumulator is not in the full capacity (maximum operation pressure). In this case, the vehicle braking is identified through a sensor located in the pedal brake module (13), that sends a signal to the air conditining module (15) and re-sends a signal to the clutch of the regenerative system (9), driving the kinetic energy transfer to the compressor. When the clutch of the transmission is in operation, the clutch of engine (10) will not be in operation. When the pressure sensor (16) identifies the maximum operation pressure, a signal is sent to the air conditioning ECU Conditioning Air (15) that forward the signal to the transmission clutch of the regenerative system (9), disactivating it.

c) Energy transfer from engine to the compressor: this operation mode occurs when the air conditioning system is turned on and the vehicle is not in braking process. In this case, air conditioning system ECU (15) receives the information from the brake pedal module (13) indicating that the vehicle is not in braking process. In parallel, the air conditioning system ECU (15) verifies if the air conditioning system is turned on: in affirmative case, the air conditioning system ECU (15) sends a signal to the engine clutch (10), allowing the kinetic energy transfer from the engine (7) to the compressor (1), through the belts (11). With this kinetic energy transfer from the engine to the compressor there is a kinetic energy deficit for the vehicle propulsion: then the air conditioning system ECU (15) sends a signal to the engine ECU (14) allowing a load compensation for the engine.

The air flow supplied to the cabin is controlled through a valve (17) located in the air line pipe between the pneumatic accumulator and the vortex tube. This valve has the function of the air compressed flow control and must be piloted, so that the air conditioning ECU can command its opening or closing as needed.

The air flow temperature control is executed through the joined control of the compressed air flow and a fan module air flow. A reduction in the compressed air flow that is supplied to the vortex tube will cause the reduction of the cold air flow. Then, it is necessary to compensate the reduction of the outflow and temperature control: an additional air flow (ambient temperature) is supplied through the module fan and mixed with the cold air flow, obtaining a final air flow in the desired temperature.

5.2. Components Selection

5.2.1. Vortex tubes

Initially, in order to establish the necessities of the system, it was necessary to obtain the thermal load. Based on an external temperature of 311K, temperature of the evaporator of 283K, relative humidity in the external environment of 40%, relative humidity in the evaporator of 100% and air flow rate of 0.1285 kg/s (Forrest and Bhatti, 2002), it was obtained a thermal load of 1.6 TR.

As presented by Stephan et al. (1983), the temperature variation of the cold air, supplied by the vortex tube, depends on the cold air flow ratio. Tab. (2) presents the temperature variation of the cold air flow as a function of cold air ratio and vortex tube operation pressure, according Itao (2005).

Table 2 – Temperature drop of the cold air flow in function of the cold air flow ratio in Vortex Tubes

Vortex tube Inlet air pressure (p_0)	**Cold air flow ratio (y_c)**						
	0.2	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**
kPa	ΔK	ΔK	ΔK	ΔK	ΔK	ΔK	ΔK
138	**34.4**	**33.3**	**31.1**	**28.3**	**24.4**	**20.0**	**15.6**
276	**48.9**	**47.2**	**44.4**	**40.6**	**35.0**	**28.9**	**21.1**
414	**57.8**	**55.6**	**51.7**	**46.7**	**40.6**	**33.3**	**25.6**
552	**63.9**	**61.1**	**56.7**	**51.1**	**44.4**	**36.7**	**27.8**
689	**68.3**	**65.6**	**61.1**	**55.6**	**47.8**	**39.4**	**30.0**
827	**71.7**	**68.9**	**64.4**	**57.8**	**50.6**	**41.1**	**30.6**

Once a high cold air flow ratios result in low temperature variations and intermediate cold air flow ratios result in higher temperature variations, it is necessary to analyze the best combination that results in the optimum performance of the system. Figure (8) presents a graph of the operation pressure versus the cold air flow ratio necessary to obtain temperature variation of 28K.

With the results presented in Fig. (8), it was calculated the necessary inlet vortex tube air flow, that is presented in Tab. (3).

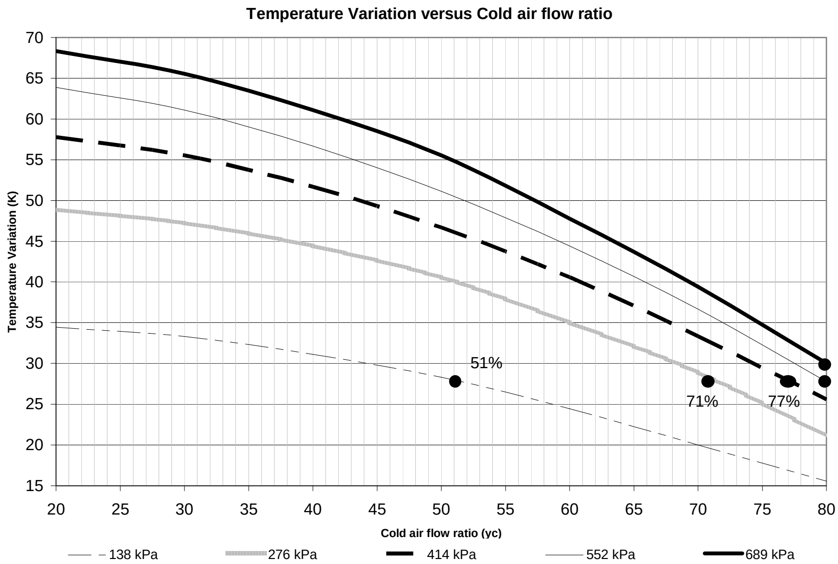


Figure 8. Variation of Temperature in function of the cold air flow ratio

Table 3 – Necessary inlet air flow for diverse vortex tubes operation pressures

Inlet vortex tube air pressure p_0 (kPa)	**Cold air flow ratio** y_c (-)	**Inlet volumetric air flow required** $\dot{V}_{req}$ (m^3/s)	**Inlet air flow rate required** $\dot{m}_{req}$ (kg/s)
138	0.51	0.176	0.216
276	0.71	0.127	0.155
414	0.77	0.117	0.143
552	0.80	0.113	0.138
689	0.80	0.113	0.138

*ρ air = 1.225 kg/m³

The selection of the vortex tube is closely related to the compressor and air reservoir. Modifying the operation pressure of the vortex tube, the required power for the air compression is also modifyed. Then, it is necessary to analyze the compression power for each condition presented in Fig. (6). Brown (1997) presents the equations for the required theoretical power calculation:

$$\dot{W}_{compressor} = \frac{p_1.\dot{V}_{req}}{\eta_{compressor}}.\frac{k}{k-1}.\left(r_p^{\left(\frac{k-1}{k}\right)} - 1 \right) \tag{6}$$

$$r_p = p_2 / p_1 \tag{7}$$

Where $\dot{W}_{compressor}$ is the theoretical required compressor power (kW), $\dot{V}_{req}$ is the inlet volumetric air flow required (m^3/s), $\eta_{compressor}$ is the compressor efficiency (-), k is the specific heat ratio (-), r_p is the pressure ratio (-), p_1 is the compressor inlet air pressure (kPa) and p_2 is the compressor outlet air pressure (kPa).

Using the data obtained in Tab. (3) and applying in Eq. (2) and Eq. (3), it was obtained the required theoretical power, presented in the Tab. (4).

Table 4 – Required power for compression in diverse operation pressures

Compressor outlet air pressure p_2 **(kPa)**	**Compressor inlet air pressure** p_1 **(kPa)**	**Cold air flow ratio** y_c **(-)**	**Inlet volumetric air flow required** $\dot{V}_{req}$ **(m^3/s)**	**Inlet air flow rate required** $\dot{m}_{req}$ **(kg/s)**	**Pressure ratio** r_p **(-)**	**Specific heat ratio[1]** k **(-)**	**Compressor efficiency[1]** $\eta_{compressor}$ **(-)**	**Theoretical required compressor power** $\dot{W}_{COMPRESSOR}$ **(kW)**
140	101.3	0.51	0.176	0.216	1.4	1.395	0.73	8.6
280	101.3	0.71	0.127	0.155	2.8	1.395	0.81	19.0
410	101.3	0.77	0.117	0.143	4.1	1.395	0.84	24.5
550	101.3	0.80	0.113	0.138	5.5	1.395	0.85	29.4
690	101.3	0.80	0.113	0.138	6.9	1.395	0.84	34.9

[1] Brown (1997)

By analyzing the Tab. (4), it can be verified that although the operation pressure 140 kPa results in the necessity of a inlet air flow higher than the other operation pressures, the required power is the lowest. This can be explained observing Eq. (6), where the required power is not proportional to the pressure ratios.

Then, the operation of the vortex tube in the opration pressure of 140 kPa and cold air flow ratio of 51%, becomes the most viable hypothesis of configuration.

It must be pointed out that it is possible to select a higher amount of vortex tubes of inferior volumetric air flow: the selection is based on the criterion of the designer, to evaluate the best proposal of cost and benefit of the system.

The air leaves the vortex tube in high speed, hence high levels of noise are common around the vortex tube outlet. To minimize this inconvenience, it can be included devices as noise mufflers, that minimize the noise, or evaluate the positioning configuration of the system in the vehicle interior.

5.2.2. Pneumatic accumulator

Compressed Air and Gas Institute (2004), indicates that the pneumatic accumulators (air reservoirs) present diverse functions, such as: to compensate sudden consumptions that the compressor cannot supply momentary, to reduce the frequent necessity of the compressor load changes, to cushion the pulsations in the discharge tubing and to condense some humidity that can be present in the compressed air. In the studied application case, the pneumatic accumulator has to store compressed air, for the compressor use in the occasions of vehicle braking, when the regenerative system operates.

Compressed Air and Gas Institute (2004) still cites an equation that relates the reservoir volume, with the necessary time until the operation pressure reach a minimum level, that affects the performance of the system:

$$t = V_{reserv}.\frac{p_{\max} - p_{\min}}{\dot{V}_{req}.p_{atm}}.60 \tag{8}$$

Where t is the discharge time (s), V_{reserv} ist the reservoir total volume (m^3), p_{max} is the maximum resevoir pressure (kPa), p_{min} is the minimum reservoir pressure (kPa) and p_{atm} is the atmospheric pressure (kPa).

The selection of the reservoir volume, as well its maximum operation pressure, is closely related to the compressor capacity. For the current case, as established before, a minimum pressure of operation (equivalent to 140 kPa) is necessary, so that the desired temperature variation is reached (28K). Moreover, when the air conditioning system operates in full load, it is necessary a volumetric outlet flow of 0.176 m^3/s. Then, with these data, it is possible to simulate the time of discharge, the maximum operation pressure and the necessary volume, using the Eq. (8). This analysis is presented in Tab. (5).

From Tab. (5), it is verified that the discharge time increases the bigger the reservoir volume or the bigger the maximum storage pressure. The existence of a long discharge time is interesting, however this causes some consequences:

a) Usually, the passenger vehicles do not have much available space. Thus, having a greater air reservoir volume would increase the mass of the vehicle minimizing its performance.

b) A higher maximum operation pressure results in a necessity of bigger compression work. So, operating in the operation mode that transfers kinetic energy from the engine to the compressor, there will be a fuel consumption increasing.

Then, based on the presented analysis, the application viability of the system for passenger vehicles is low, mainly because of the available space in the vehicle.

The application viability is greater in commercial vehicles that have a bigger available space. For this application, a proposal is the use of a 0.8 m^3 air reservoir with a maximum operation pressure of 800kPa, that it constitutes a half term between available space and pressure of operation. Consulting the Tab. (5), it can be verified that this configuration requires a discharge time of 30s. This means that the air conditioning system can operate in maximum load (without the compressor driving) during 30s, if the reservoir pressure in the operation beggining is equivalent to 800 kPa.

Table 5 – Discharging time as a function of the reservoir volume for diverse operation pressures

Reservoir total volume V_{reserv} (m^3)	**Maximum reservoir pressure** p_{max} (kPa)	**Minimum reservoir pressure** p_{min} (kPa)	**Inlet volumetric air flow required** $\dot{V}_{req}$ (m^3/s)	**Atmospheric pressure** p_{atm} (kPa)	**Discharge Time** t (s)	**Discharge Time** t (min)	**Reservoir total volume** V_{reserv} (m^3)	**Maximum reservoir pressure** p_{max} (kPa)	**Minimum reservoir pressure** p_{min} (kPa)	**Inlet volumetric air flow required** $\dot{V}_{req}$ (m^3/s)	**Atmospheric pressure** p_{atm} (kPa)	**Discharge Time** t (s)	**Discharge Time** t (min)
2.0	200	140	0.176	101.3	7	**0.11**	0.8	200	140	0.176	101.3	3	**0.04**
2.0	500	140	0.176	101.3	40	**0.67**	0.8	500	140	0.176	101.3	16	**0.27**
2.0	800	140	0.176	101.3	74	**1.23**	0.8	800	140	0.176	101.3	30	**0.49**
2.0	1000	140	0.176	101.3	96	**1.61**	0.8	1000	140	0.176	101.3	39	**0.64**
2.0	2000	140	0.176	101.3	209	**3.48**	0.8	2000	140	0.176	101.3	83	**1.39**
1.0	200	140	0.176	101.3	3	**0.06**	0.5	200	140	0.176	101.3	2	**0.03**
1.0	500	140	0.176	101.3	20	**0.34**	0.5	500	140	0.176	101.3	10	**0.17**
1.0	800	140	0.176	101.3	37	**0.62**	0.5	800	140	0.176	101.3	19	**0.31**
1.0	1000	140	0.176	101.3	48	**0.80**	0.5	1000	140	0.176	101.3	24	**0.40**
1.0	2000	140	0.176	101.3	104	**1.74**	0.5	2000	140	0.176	101.3	52	**0.87**

5.2.3. Air compressor

Nowadays, there are many compressor types, available in the market. Compressed Air and Gas Institute (2004) cites several of them. Their uses are related to some variables, as the discharge pressure and admission capacity.

In the operation mode of the regenerative system, the compressor must operate until a maximum pressure of 800 kPa and a maximum volumetric air flow of 0.176 m^3/s. In order to obtain the necessary power for the maximum pressure of operation, it was applyed the Eq. (6). The results are presented in Tab. (6).

Table 6 – Necessary power in function of the operation pressure

Compressed outlet air pressure p_2 **(kPa)**	**Compressed inlet air pressure** p_1 **(kPa)**	**Inlet volumetric air flow required** $\dot{V}_{req}$ **(m^3/s)**	**Inlet air flow rate required** $\dot{m}_{req}$ **(kg/s)**	**Pressure ratio** r_p **(-)**	**Specific heat ratio**[1] **k (-)**	**Compressor efficiency**[1] $\eta_{compressor}$ **(-)**	**Theoretical required compressor power** $\dot{W}_{COMPRESSOR}$ **(kW)**
200	101.3	0.176	0.216	2.0	1.395	0.73	18.8
300	101.3	0.176	0.216	3.0	1.395	0.73	31.6
400	101.3	0.176	0.216	4.0	1.395	0.73	41.6
500	101.3	0.176	0.216	5.0	1.395	0.73	49.9
600	101.3	0.176	0.216	6.0	1.395	0.73	57.2
700	101.3	0.176	0.216	7.0	1.395	0.73	63.6
800	101.3	0.176	0.216	8.0	1.395	0.73	69.4

[1] Brown (1997)

Analyzing the Tab. (6), it can be verified the necessity of high compression power, for the outlet air pressure conditions. This demonstrates the necessity of high regeneration powers.

It is also necessary to point out the necessity of a heat exchanger that minimizes the air temperature in the compressor outlet. The air, as cites Compressed Air and Gas Institute (2004), suffers a temperature rise during the compression process. A hot air inlet in the vortex tube must be prevented, because the cold air flow temperature will be higher in this condition.

5.3. Viability analysis (point of view of energy regeneration)

The energy regeneration data for commercial vehicles and passenger vehicles in 2 operation cycles have already been presented previously. In order to compare the regeneration data with the energy needs for compression, it was created Tab. (7).

Comparing the necessary compression power with the available regeneration power, it can be concluded that by the point of view of compression power, it is viable only the application in commercial vehicles (mainly in the condition with trailer). Although, even in commercial vehicle without a trailer, it is possible to obtain the necessary power to supply the maximum compression power.

In the case of commercial vehicle with trailer, because of the higher available power for regeneration, compressors of higher capacity could be used optimizing the air volume storage. However, the increase of volume and weight of the compressor must be carefully considered: these variables can invalidate the project.

Table 7 – Available regeneration power versus necessary compression power

Vehicle	Cycle	Time (s)	Available energy (kJ)	Available power for regeneration (kW)	Maximum required compression power (kW)
Passenger vehicle	NBR 6601	1370	1234	~ 0.90	69.4
Passenger vehicle	40-0 km/h	5	52	~ 10.4	69.4
Commercial vehicle without trailer (15ton.)	40-0 km/h	5	413	~ 82.6	69.4
Commercial vehicle with trailer (15ton.)	40-0 km/h	5	1337	~ 267.4	69.4

Adopting the hypothesis that a conventional air conditioning system needs a minimum power of 3.0 kW for its operation, the fuel consumption economy for a commercial vehicle (engine power equivalent to 200kW) using the concept of energy regeneration, can exceed 5%. This can be possible if the regenerative system also integrates the pneumatic system for the air brakes and suspension.

6. Conclusions

Since prior projects of automotive air conditioning systems, many advances have been reached optimizing the functioning and the efficiency of such system. However, there are still some difficulties present nowadays (for an example the difficulty of homogeneous climatization in the vehicle cabin).

Then the research of alternative climatization ways is interesting and necessary. A possibility that was studied in this work is the application of the vortex tube to the vehicle climatization. To the vortex tube operation, it was additioned an air compression system, that can reutilize the movement of the vehicle axle during the braking process, storing compressed air for future use. Moreover, it was foreseen the operation of the air compressor with kinetic energy proceeding from the internal combustion engine. This happens when there is not possibility of the regenerative system operation. Such configuration philosophy allows the possibility of the vortex tube operation, even in the condition of engine off. In this case, the system operates until the air reservoir reachs a minimum operation pressure.

Based on the sizing data of conventional air conditioning systems for passenger vehicles, it was obtained a necessary thermal load, equivalent to 1.6 TR. Utilizing the data of a cold air flow of 0.090 m^3/s and a temperature differential of 28K, it was executed a first-sizing of the operation pressure and of the cold air ratio (important variables for the vortex tube performance). The results have shown a minimum air compression power consumption, when using a cold air flow ratio of 51% and an operation pressure of 140 kPa.

As mentioned previously, the operation pressure can determine the viability or not of the system. A high operation pressure demands a high air compression power. The results have shown that a low operation pressure of the vortex tube, even with the necessity of the increasing in the compressed air flow supplying, makes possible the necessity of a lower compression power (8.6 kW).

The comparison between the regeneration power data and compression power data has shown the viability of the application of the system only for commercial vehicles. There are many limitations for the application in passenger vehicles.

The main limitation is related to the lower available energy for regeneration, that is not enough for the vortex tube air conditioning system driving.

The second limitation is related to the air reservoir. For the studied system, it was selected a compressed air reservoir volume of 0.8 m^3. Converting this volume in linear dimensions, it would be necessary about 4 air reservoirs with 0.60m of diameter and 0.70m of length, making impracticable the application of the system for passenger vehicles, due to the space and weight limitation.

The third limitation is related to the compressor. The application of an air compressor, that can compress a high air flow (0.176 m^3/s),is impracticable in passenger vehicles due to the space and weight limitation.

In commercial vehicles, the viability application is higher, due to higher mass and the higher available space. If the commercial vehicle is equipped with a braking pneumatic system and/or an air suspension, the air conditioning system with regeneration could be integrated with the pneumatic system in order to optimize the project.

The sizing of the system that controls the transfer of the kinetic energy from vehicle axle to the compressor must be carefully executed, so that the deceleration does not impair the control of the vehicle. For commercial vehicles that have superior mass (commercial vehicle with tow) this effect is lower, however must be taken into account.

The present work did not deal with the complete sizing of all the components, because it was not the objective of this study. Then, it is suggested for future studies of this subject, the analysis and detailed sizing of the components, as well the energy regenerative system control.

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