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## Capacity and Acoustic Optimization on a Fan/Heat Exchanger Unit

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### ABSTRACT

In this paper, an optimization was done on a fan / heat exchanger unit. The optimization software contains models for the heat transfer and pressure loss in the condenser and an acoustic model for the fan. The models were based on fan and heat exchanger dimensions (like length, height, width and number of fins). We are interested in this type of study in order to find the best combination of fan acoustic parameters and heat exchanger capacity.

The heat exchanger is defined using a logarithmic measurement of temperature differential (DTLM) model. The heat transfer coefficient is based on the Nusselt number for the refrigerant and the Colburn factor for the air using the geometric characteristics of the heat exchanger. The fan acoustic model is related to the airflow rate defined using the velocity, diameter, flow rate and acoustic level of the existing fan.

The optimization solver iterates on the parameters of the heat exchanger and fan in order to give the best combination. The optimized result showed a 5dBA improvement over the base line configuration.

The optimization showed that the most influential parameters for heat exchanger efficiency are height, width, length and number of fins. However, some of these parameters tend to increase the pressure loss, which decreases the airflow rate, and influences the acoustic level. Using the method described here, we were able to determine the geometric arrangement that optimizes the efficiency and acoustic objectives.

### 1. INTRODUCTION

Today, the acoustic level of Condensing Unit application is important for user, that why I focus on the coupling between the fan and heat exchanger. The acoustic levels of the coupling depend of the fan velocity and geometric parameter which have an impact on the heat exchanger capacity. That why, I decide to create thanks to an Optimization software an algorithm based on analytic model of the coupling of fan/ heat exchanger in order to define the best combination. The main objective of this subject is to develop a tool allowing to predict the performance of the coupling in reducing the development time and help designer to create well at the first time.

The first part of the present paper is devoted to a description of the analytic model on the coupling between the fan and heat exchanger integrating geometric parameter like diameter of the fan, number of tubes, length, width, height, fins of the heat exchanger.

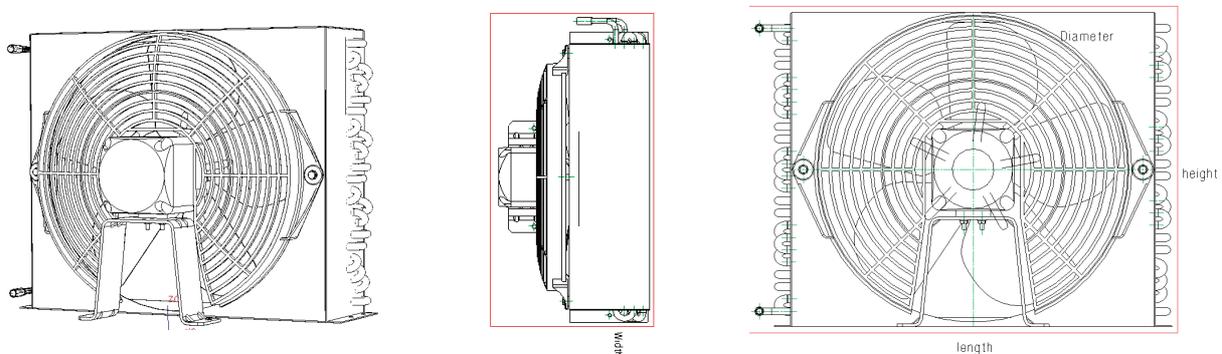


Figure1: Heat exchanger and fan system on the Condensing Unit

This model can define the best geometric arrangement according the acoustic level objective and other objective thanks the heat exchanger capacity and the price of the coupling. The validation of this model is based on the test and technical database from heat exchanger supplier which is described on the second part of the paper. Testing optimization tool, we can bring out parameters allowing to improve the existing coupling between fan and heat exchanger in order to change part and obtain the best acoustic level fan and heat exchanger capacity. At the end of the paper, a demonstration on the utilization of the optimization tool will be illustrated on the CAJ9480ZMHR condensing Unit.

## 2. Analytic model on the coupling between fan and heat exchanger

The definition of heat exchanger capacity and fan acoustic behavior require the knowledge of the inlet parameter like temperatures for heat exchanger and air velocity for the fan. The inlet data define the choice of the analytic model methodology to use for optimization. The model of the coupling will be based on the geometric parameter of the parts in order to know the impact on the acoustic performance of the fan without damage the heat exchanger capacity and the price. On this part of the paper, I describe the mathematical expression using to develop the algorithm code on the optimization software.

### 2.1 Heat exchanger model

2.1.1: heat exchanger capacity : In this case of study, the prediction of the heat exchanger capacity will be define thanks the logarithmic measurement of temperature differential (DTLM) methodology instead of the transfer unit number (NUT) methodology by reason of the input data that I can use : input and out put temperature only. Thus, the general definition expression of the heat exchanger is like following:

$$Q_k = K.S.DTLM \tag{1}$$

On this expression, the surface (S) is defined according to the combination thanks to the following parameters:

- Number of fin: Nfin
- Number of tube: Ntube
- Height of heat exchanger: H
- Width of heat exchanger: W
- Length of heat exchanger: l
- Inner and outer of tube diameter: Di & De
- And the fin thickness: e

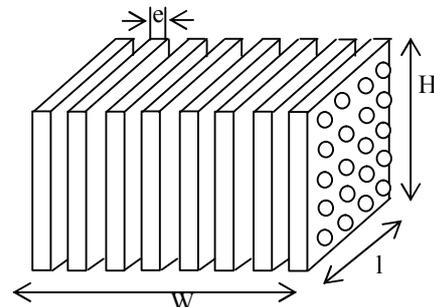


Figure 2 : heat exchanger draft

$$S = \left[ (2 \times ((l \times H) + e \times (l + H)) - ((\pi \times \frac{D_i^2}{4}) \times Ntube)) \times Nfin \right] + [W \times De \times Ntube] \tag{2}$$

The surface expression is made of fin and tubes surface according to the heat exchanger parameters. This expression gives similar result in comparison with the heat exchanger supplier data. The following step of the model is to define the DTLM parameter according the temperatures of fluids, like following expression:

$$DTLM = \frac{(T_{sf} - T_{eair}) - (T_{ef} - T_{sair})}{\log \left( \frac{T_{sf} - T_{eair}}{T_{ef} - T_{sair}} \right)} \tag{3}$$

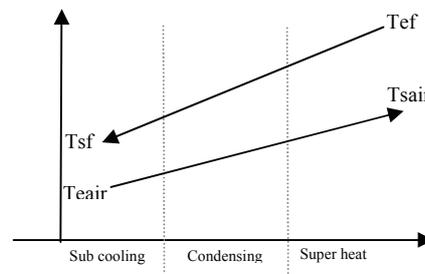


Figure 3: Temperature profile of the fluids through the heat exchanger

On the DTLM definition, I consider the super heat and sub cooling phenomenon on the heat exchanger in order to reduce the gap between simulation and test result. After that, I integrate the heat coefficient “K” defined by the air flow and refrigerant like the following expression:

$$\frac{1}{K} = \left( \frac{1}{h_{ff}} \right) + \frac{1}{h_{air} \cdot \frac{Se}{Si} \cdot (\eta_{eff} \cdot S_{fin} + 1)} + \frac{Di}{2 \cdot \lambda_{wall}} \cdot \log \frac{De}{Di} \quad (4)$$

The definition of exchange ratio by air (hair) and refrigerant (hff) on this expression has been defined according to the Colburn correlation for air and Nusselt number for refrigerant. On the model, I consider a condenser with smooth fin and circular tubes used on dry condition. Thus, the exchange ratio on air side is defined by Gray and Webb proposition for Colburn factor like following:

$$J_{air} = 0.14 \cdot Re^{-0.328} \cdot \left( \frac{St}{Sl} \right)^{-0.5} \cdot \left( \frac{p}{De} \right)^{0.31} \quad (5)$$

And the exchange ratio on refrigerant side is defined according the Nusselt number like following:

$$Nu = \alpha \cdot Re_{fluid}^{\beta} \cdot Pt_{fluid}^{\gamma} \quad (6)$$

$$\alpha = 0,33 \quad ; \beta = 0,55 \quad ; \gamma = 1/3$$

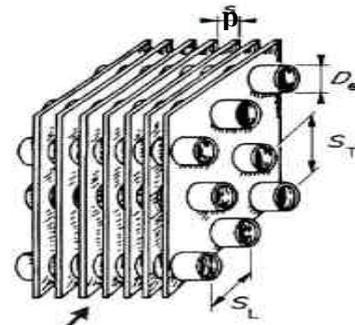


Figure 4: The type of heat exchanger on the model

On the code development, the heat exchanger is the most complicate model to make on the coupling. Indeed, on the heat exchanger the flow is on two phases (liquid and gas) which mean different thermodynamics characteristics. In spite of the definition by correlations, I adapted the coefficient number of the Nusselt according to the test result in order to obtain less than 10% of error.

2.1.1: heat exchanger pressure loss: The airflow goes through the heat exchanger create pressure loss according to geometry. The pressure loss is composed by singularity like restriction which can define like following expression:

$$\Delta P_{condo} = \xi \times \frac{1}{2} \times \rho_{air} \times v_{air}^2 \quad (7)$$

$$\xi = 3.6 \times (Kc + Ke + 4 \times f_{wall} \times \left( \frac{1}{Dh} \right)) \quad (8)$$

The  $\xi$  factor on the pressure loss expression is composed of Kc & Ke: singularity expression,  $f_{wall}$  represent the rubbing phenomenon between the airflow and the fin .The Dh parameter is the surface crossing by air on the heat exchanger. Each parameter of  $\xi$  is defined according for the following expressions:

$$Kc = 0.08 \times (1 - \sigma^2) \quad (9)$$

$$Ke = 0.2 \times (1 - \sigma^2) \quad (9)$$

$$\sigma = 1 - \left( \frac{Nfin \times e}{W} \right) \quad (11)$$

$$f_{wall} = \frac{\left[ \left( \frac{3.44}{\sqrt{L}} \right)^2 + (f \times Re)^2 \right]^{\frac{1}{2}}}{Re} \quad (13)$$

$$L = \frac{l / Dh}{Re} \quad (14)$$

The heat exchanger capacity and pressure loss have been verified by test in order to evaluate the reliability of the model which the comparisons of results have been presented on the following paragraph. The link between the heat exchanger and the fan is done by the airflow rate used also to define the heat exchanger capacity and pressure loss.

**2.1 Fan acoustic model**

The analytic model of the fan is based on the acoustic similitude law describing the fan behavior. This model could define the characteristic of new fan (2) according to the existing fan (1). However, this law will be able if the ratio of the diameters (D1/D2) is lower or equal to 3.

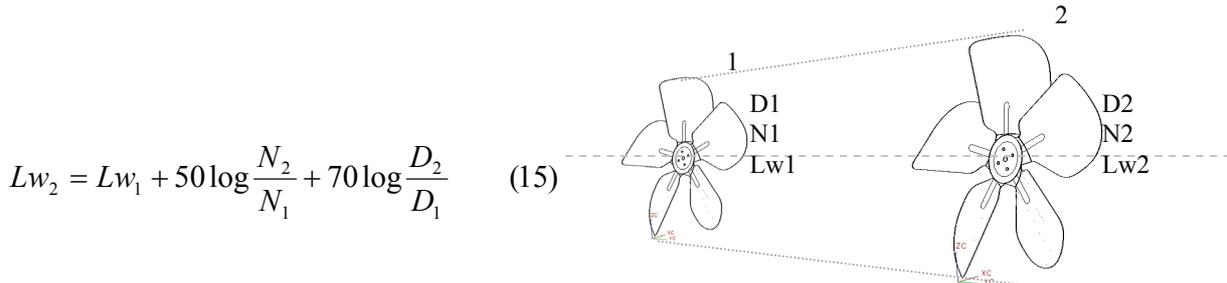


Figure 5: Fans models

To respect the ratio between the diameters, I set a relation between the heat exchanger height and the diameter of the new fan in order to respect the value of homothetic ratio on D1/D2. The definition below of acoustic behavior of fan satisfy to link with heat exchanger capacity and pressure losses. Thus, the acoustic level of the coupling will be defined thanks to the airflow rate parameter existing on the both model.

**2.1 The coupling model**

Base on the airflow rate, the coupling need the fan pressure losses curve in order to realize the optimization. Indeed, the airflow rate take an idea on the heat exchanger capacity and that is the single parameter can link the heat exchanger and the acoustic models' of fan. Thus, after the definition of the pressure losses of the heat exchanger and take the references of the fan curve on the optimization software, the coupling model code is define like the following expression:

$$\Delta P_{Qk} - \Delta P_{fan} = 0 \quad (16)$$

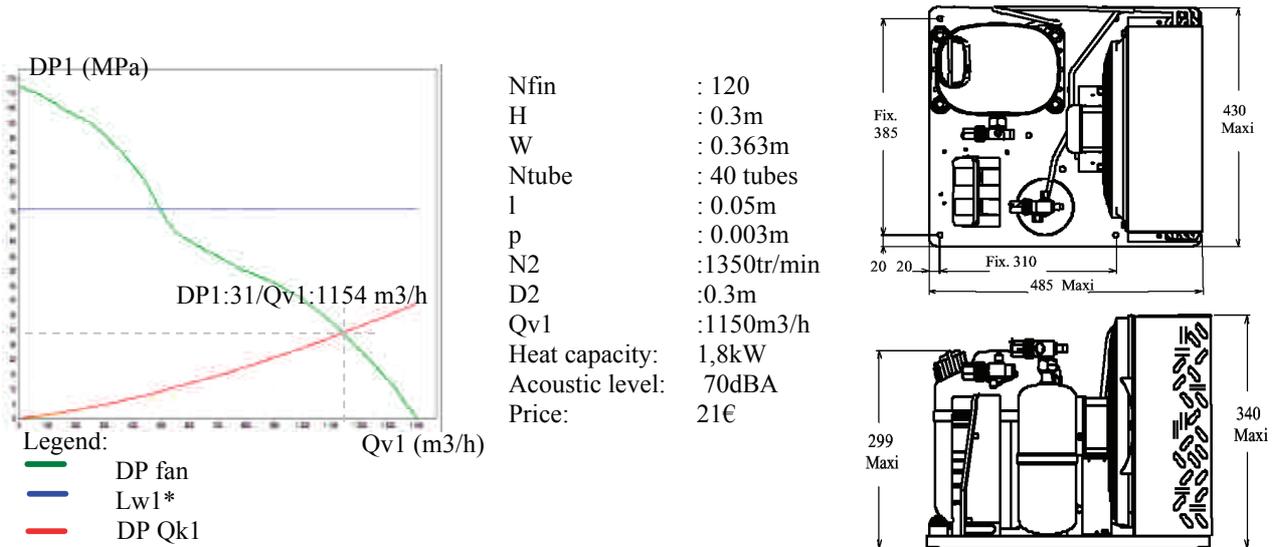


Figure 6: The characteristic of the coupling models' of CAJ9480ZMHR condensing unit

### 3. MODEL VALIDATION

The validation of the heat exchanger model is based on test. The tests have been useful to determine the exchanger coefficient ratio on refrigerant side. The matter of the test is to modify the velocity of the fan in order to evaluate the evolution of the heat capacity. The test result could estimate the reliability of the algorithm model of the coupling and to fit the Nusselt coefficient.

#### 3.1 Heat exchanger validation

The boundary condition of the test is like following:

Let the cooler temperature fix, and change only the fan velocity to modify the airflow.

The boundary condition of the supplier data base is like following:

Let the heat exchanger geometry fix, and change only the airflow data.

The boundary condition of the simulation is like following:

Let the heat exchanger geometry fix, change the airflow rate and define thermodynamics properties of the refrigerant (temperature and enthalpy) according the airflow.

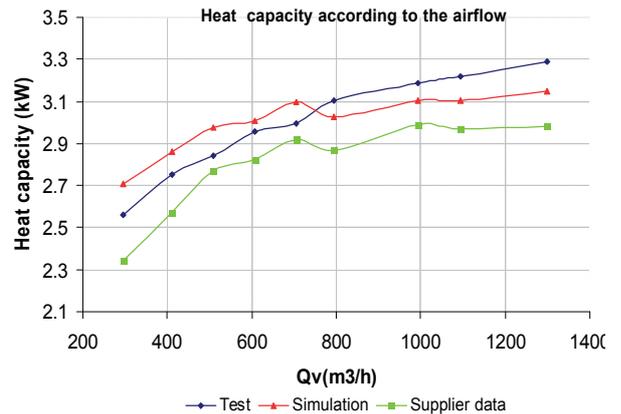


Figure 7: Comparison on test and simulation of the heat capacity according to the airflow

The experimental set-up consists of a condensing unit mounting with manometer on the heat exchanger surface (see figure8), in order to estimate the airflow according to the fan velocity. Thus, the airflow measurement is over estimated by this experimental methodology. That why, we can notice some differences at the end of the heat capacity curve (see figure7) between the simulation and test.

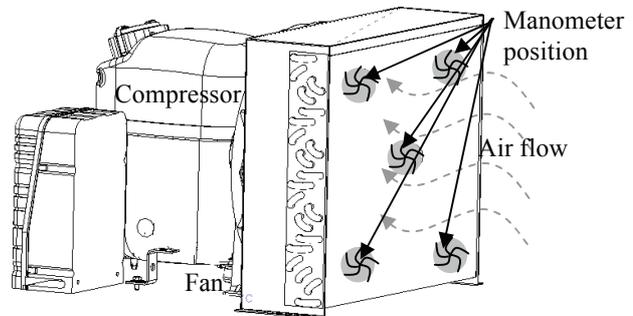
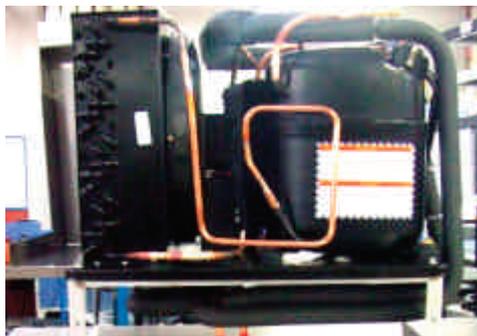


Figure 8: Experimental set up

However, the test and supplier data base could adjust the Nusselt coefficient in order to reduce the gap with the test result less than 10% of error. Thus, the new Nusselt expression is like following:

$$Nu = \alpha \cdot Re_{fluid}^{\beta} \cdot Pt_{fluid}^{\gamma} \quad (17)$$

$$\alpha = 1,06 \quad ; \beta = 0,51 \quad ; \gamma = 1/3$$

### 4. OPTIMIZATION

The optimization is based on existing condensing unit call CAJ9480ZMHR (see characteristic on figure 6). The optimization must satisfy the global dimension, respecting any objectives like heat capacity, acoustic level and the price. That why, the model of the coupling consider geometric aspect of the fan and heat exchanger. Thus, the optimization will be done after the definition of the constraints.

#### 4.1 Optimization result according to the objective:

Before, to start the optimization, the constraints which must be respected in the case of this study, is like following:

Constraints about the heat exchanger are:

- Number of fin:  $90 < N_{fin} < 130$
- Number of tube:  $30 < N_{tube} < 50$
- Height:  $0,2 < H < 0,35$
- Length:  $0,2 < W < 0,42$
- Width:  $0,03 < l < 0,18$
- Sl :  $Sl = 0,0125$
- St :  $St = 0,025$

Constraints about the fan are:

- Velocity ratios:  $0 < N1/N2 < 3$
- Diameter ratios:  $0 < D1/D2 < 3$
- Velocity on the base configuration:  $N1 = 1350 \text{ tr/min}$
- Acoustic on the base configuration :  $Lw1 = 70 \text{ dBA}$
- Airflow on the base configuration:  $Qv1 = 1130 \text{ m}^3/\text{h}$

Other constraints will be take, like  $D2 < H$  or  $Qv1 < Qv2$ , in order to obtain good result after the iteration of the software. All of the constraints above take on the optimization software.

4.1.1: Optimization result according to the heat capacity objective: In the case of this optimization, the result could give the best geometric combination to have the height heat capacity of the coupling.

For this objective the best combination is:

- Nfin : 130
- H : 0.35m
- W : 0.409m
- Ntube : 48 tubes
- l : 0.057m
- p : 0.003m
- N2 : 1709tr/min
- D2 : 0.31m

The result of the optimization is:

- Heat capacity: 2,78kW
- Acoustic level: 75dBA
- Price: 26,9€

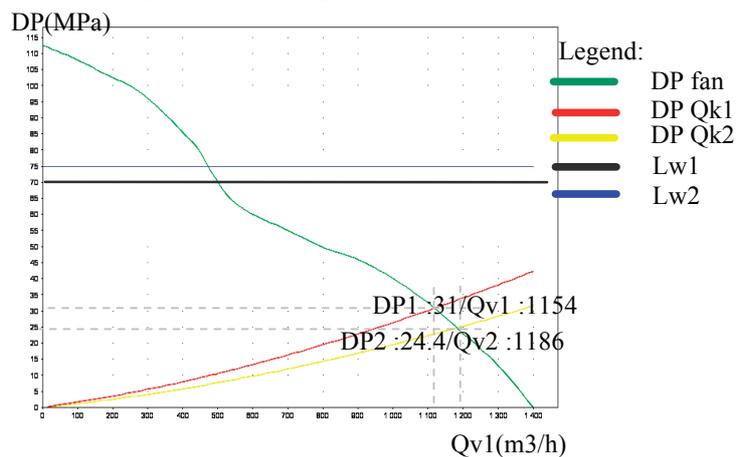


Figure 9: Coupling point for heat capacity objective

4.1.2: Optimization result according to the acoustic objective: In the case of this optimization, the result could give the best geometric combination to have the less acoustic level of the coupling.

For this objective the best combination is:

- Nfin : 120
- H : 0.35m
- W : 0.42m
- Ntube : 40 tubes
- l : 0.05m
- p : 0.003m
- N2 : 1240tr/min
- D2 : 0.32m

The result of the optimization is:

- Heat capacity: 2,1kW
- Acoustic level: 64dBA
- Price: 22,6€

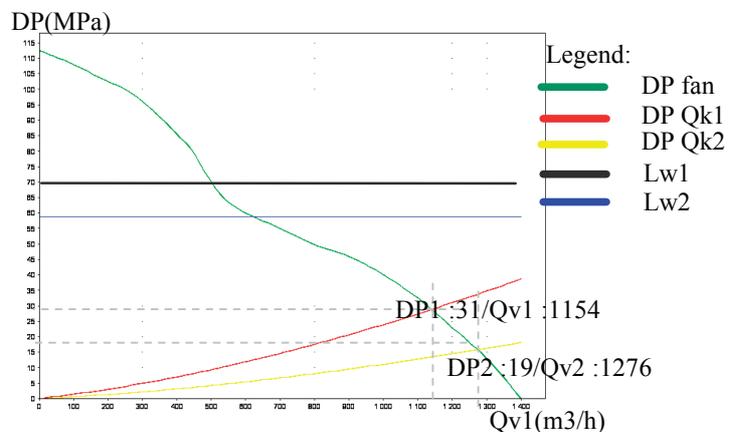


Figure 10: Coupling point for acoustic objective

4.1.3: Optimization result according to the coupling price objective: In the case of this optimization, the result could give the best geometric combination to have the less price of the coupling.

For this objective the best combination is:

Nfin : 100  
 H : 0.3m  
 W : 0.295m  
 Ntube : 30 tubes  
 l : 0.05m  
 p : 0.0028m  
 N2 : 2100tr/min  
 D2 : 0.27m

The result of the optimization is:

Heat capacity: 1.3kW  
 Acoustic level: 78dBA  
 Price: 11.94€

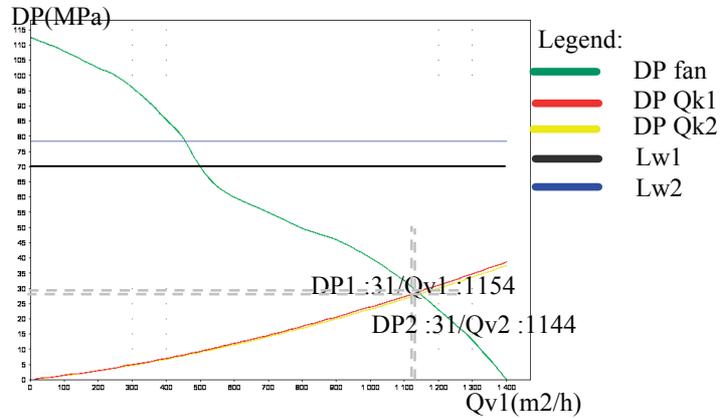


Figure 11: Coupling point for price objective

4.1.3: Optimization result according to the all objectives: In the case of this optimization, the result could give the best geometric combination to have the less price and acoustic level with the height heat capacity of the coupling. It is not possible to do multi objectives optimizations with the software, that why, I take new constraints in order to obtain the best combination according the all objective. The optimization above can define the new constraint to respect the all objectives. Thus, I add the following constraint:  $Lw2 < 65dBA$  and  $Qk > 1.9kW$  minimizing the price.

For these objectives the best combination is:

Nfin : 128  
 H : 0.35m  
 W : 0.412m  
 Ntube : 30 tubes  
 l : 0.057m  
 p : 0.003m  
 N2 : 1277tr/min  
 D2 : 0.32m

The result of the optimization is:

Heat capacity: 2kW  
 Acoustic level: 65dBA  
 Price: 18,4€

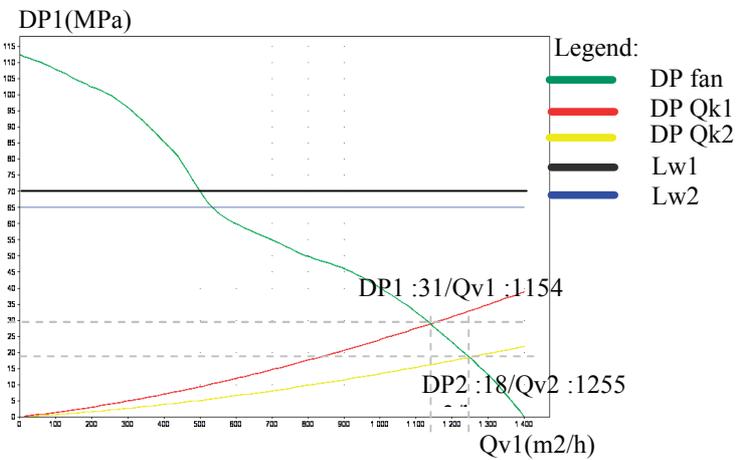


Figure 12: Coupling point for price objective

The optimization result bring out that the CAJ9480ZMHR condensing unit will be able to improve the acoustic level without damage the capacity and price of the heat exchanger. In addition, thanks of the any optimizations, we can define the geometric part which have the most influential parameters for heat exchanger capacity and the acoustic. Indeed, The height of the condenser have an impact on the three objective. However, to improve the acoustic level, it is important to reduce the velocity of fan. The number of tubes on the condenser increase the price whereas the width can improve the heat capacity. Thanks of the algorithm, the best coupling will be done.

## 5. CONCLUSIONS

The algorithm model of the coupling for the optimization was developed to improve the existing configuration on condensing unit. An evaluation was made on the condensing unit call CAJ9480ZMHR, and the optimization bring out the possibility to improve the acoustic level. The optimization will be done for the other condensing unit, but the difficulties is to integrate automatically the thermodynamics properties calculation of the refrigerant according the condition.

This development could bring out the geometrics factors which take an impact on the heat exchanger capacity, and the acoustic level of the coupling. Thus, the improvement of the capacity need to increase the height and the number of tubes of the heat exchanger. The acoustic aspect of the coupling is only related on fan characteristics like velocity and the diameter of the fan.

## NOMENCLATURE

K	heat coefficient	(W.m <sup>-2</sup> .K)	? P <sub>Qk</sub>	condenser pressure drop	(MPa)
S	total surface of exchanger	(m <sup>2</sup> )	? P <sub>fan</sub>	fan pressure drop	(MPa)
N <sub>fin</sub>	Number of fin	(-)	Q <sub>v1</sub>	airflow of existing fan	(m <sup>3</sup> /h)
N <sub>tube</sub>	Number of tube	(-)	Q <sub>v</sub>	airflow of new fan	(m <sup>3</sup> /h)
H	height	(m)	p	not of fin	(m)
W	Width	(m)	St	transversal not	(m)
l	length	(m)	Sl	longitudinal not	(m)
Di	inner diameter of tube	(m)	Se	external surface	(m <sup>2</sup> )
De	outer diameter of tube	(m)	Si	internal surface	(m <sup>2</sup> )
e	thickness of fin	(m)	? <sub>wall</sub>	thermal conductivity of tube	(W/m.K)
T <sub>sf</sub>	outlet refrigerant temperature	(°C)	S <sub>fin</sub>	Surface of fin	(m <sup>2</sup> )
T <sub>ef</sub>	inlet refrigerant temperature	(°C)	? <sub>fin</sub>	fin efficiency	(-)
T <sub>sair</sub>	outlet temperature of air	(°C)	hair	exchange ratio by air	(W.m <sup>-2</sup> .K)
T <sub>ear</sub>	inlet temperature of air	(°C)	hff	exchange ratio by refrigerant	(W.m <sup>-2</sup> .K)
Lw1	existing acoustic level	(dBA)	? <sub>air</sub>	air density	(Kg/m <sup>3</sup> )
N1	existing velocity of fan	(m <sup>2</sup> /s)	vair	air velocity	(m <sup>2</sup> /s)
D1	existing diameter of fan	(m)	Dh	hydraulic diameter	(m)
Lw2	new acoustic level	(dBA)	f <sub>wall</sub>	factor of wall friction	(-)
N2	new velocity of fan	(m <sup>2</sup> /s)	f	factor of friction	(-)
D2	new diameter of fan	(m)	Re	Reynolds number	(-)
Pt	Prandtl number	(-)			

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