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PRESSURE STUDY OF T-SECTIONS IN HVAC SYSTEMS

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Thank you all
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Abstract

The aim of this work is to study the changes in pressure in T-sections in junctions of air conditioning systems, and the study of classical designs of flow ducts as well as the equations and theories used to obtain the major and minor pressure losses in junctions. The objective is to understand the Z factor in different conditions with the already existing ones in the ASHRAE tables.

In this work we have designed a HVAC (Heating, Ventilation, Air Conditioning) system with 90° angled tees with typical criteria for a real life situation in realistic conditions; including different sizing for the main duct and its respective branches. After determining the operation conditions of our HVAC, by means of an equation solver such as EES, we are able to calculate the different diameters required for our system in a pre-design. Once we have done our pre-design, we are able to obtain the normalized diameters of the ducts. With these, we design a realistic system using real ducts, taking into account the data obtained in the pre-design.

One of the main purpose of this study is to visualize the pressure drop throughout the system and how both fittings and friction contribute.

List of figures, equations, tables and graphs

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1. INTRODUCTION

1.1. Context

Heat, Ventilation and Air Conditioning Systems (HVAC) are typically found throughout a wide variety of buildings from different contexts and applications. They depend significantly on the infrastructure; for example, it is not the same case for hospital or a factory. A different design will be requested.

The design for various HVAC will be different but the main variables and applied methodology needed will be kept constant. The designing process will initially derive from the building requirements; such as the distribution of the different zones or rooms that are part of the system and the demands on air flows and temperature of these zones. There are some concerns upon the design state due to the complexity of some variables involved, being the main issue of them all the differences in drop pressures throughout the system. This is caused by the minor loss coefficient, also known as the Z-factor.

1.2. Objectives

Thus, the present project is a combination of a design method and a research related work on Z-factors. Onwards, we will try to achieve the following goals:

- To comprehend the performance of a HVAC system and its behaviour depending mainly on the key variables that influence on the pressure variation.
- The relevance of the study of the Z-factor and why it is determinant in a HVAC system.
- To understand, inside a particular HVAC system, the nature of T-sections as well as its variation of demeanour between the different tees.

1.3. Problem Statement

During the design process, one of the most influencing issues that have to considered are the pressure drops induced in each section. We can differentiate pressure drops due to friction, which depends on the nature of the duct, and pressure drops due to fittings, which depends on the geometry of the different sections. In our case of study, the particular shapes that we will focus on are the sections in tee. This is due to the effect that they have on the different pressures in the system.

Owing to the changing of directions between the ducts and the corresponding distribution of air flows the pressures experience a remarkable variation. These drop pressures are mainly provoked by the value of the Z-factor, which is very variable due to that a low shift in the parameters that determine it causes a disparity in the value of this coefficient.
The only reliable source of this values are the ASHRAE handbook, which is based on actual experiments. Thus, we would like to determine if there is any standard for this drop pressures in a specific tee section and if we can trust in these given values.

2. THEORETICAL BASIS

2.1. Bernouilli's Principle

For an incompressible fluid, flowing in a continuous stream, the total energy of a particle remains the same, while the particle moves from one point to another. This statement occurs assuming that there are no losses due to friction or fittings in the duct.

In a real life situation, systems are not perfect. Energy will be lost as friction and due to changes in the shape or section of the duct, that is to say, the fittings.

- **Major Loss:** The overall head loss for a duct system consists of the head loss due to viscous effects in the straight ducts.
- **Minor Loss:** The head loss in various duct components, fittings.

The equation of Bernoulli (*Equation 2.1*) can be used for incompressible fluids. Although air is not an incompressible fluid, it is incompressible in this project because of we can assume that the density is constant throughout the system as it is very hard to determine an exact value for each section. The density could be determined for each section using Computational Fluid Dynamics (CFD), but it is not the case.

Bernouilli's equation: $P_1 + \rho g l_1 + \frac{1}{2}\rho v_1^2 = P_2 + \rho g l_2 + \frac{1}{2}\rho v_2^2 + \Delta P_{fr} + \Delta P_f$ \qquad (2.1)

P: Pressure, in Pa
ρ: Fluid Density, in kg/m^3
g: Gravitational Acceleration of 9.81 m/s^2
v: Velocity, in m/s
l: Length, in m
ΔP_{fr}: Pressure Drop due to Friction
ΔP_f: Pressure Drop due to Fittings

2.2. Pressures

Pressure is the continuous physical force exerted on or against a fluid, such as air, in contact with it. The concept of pressure is vital for the study of fluids. For every point in a body of fluid, a pressure can be identified.

In fluid dynamics, we define two types of pressure which arise from Bernoulli's equation, static pressure and dynamic pressure. Bernoulli's equation is fundamental for the study of incompressible fluids and it can be directly related to pressure when the elevation term is ignored. There is a static, dynamic and total pressure for every point in a flowing fluid disregarding of the fluid velocity at that point.

$$P + \frac{1}{2}\rho v^2 = P_0 \qquad (2.2)$$

P : Static Pressure
$\frac{1}{2}\rho v^2$: Dynamic Pressure
P_0 : Total Pressure, which is constant along any streamline

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- **Static Pressure** is defined as the pressure at a point on a body moving with the fluid.

- **Dynamic pressure** is the kinetic energy per unit volume of a fluid particle.

2.3. Z-factor

Pressure drops in straight pipes or ducts are called major loss, which are a linear loss. On the other hand, pressure loss in components like valves, bends or tees are called minor loss, which are a local loss due to fittings.

The local pressure losses corresponding by fittings in the duct network are expressed *Equation 2.3*:

$$\Delta P_f = Z \cdot \frac{\rho \cdot v^2}{2} \tag{2.3}$$

ΔP_f: Pressure Drop due to Fittings
$\frac{1}{2}\rho v^2$: Dynamic Pressure
Z: Z-factor

The Z-factor is the minor loss coefficient and it is a constant which is dependent on the type of fitting. The Z-factor can be found as K, ξ, or ς in different nomenclature. According to the tables given by the ASHRAE handbook, both flow rates and areas of the different sections in the tee are needed to obtain the Z-factor. Therefore, the Z-factor in a tee will depend on the different flow rates and areas of the composing sections of the tee. For example, *Figure 2.1* shows a diverging cross, that could be also studied as a double tee, and its necessary parameters to obtain the Z-factor:
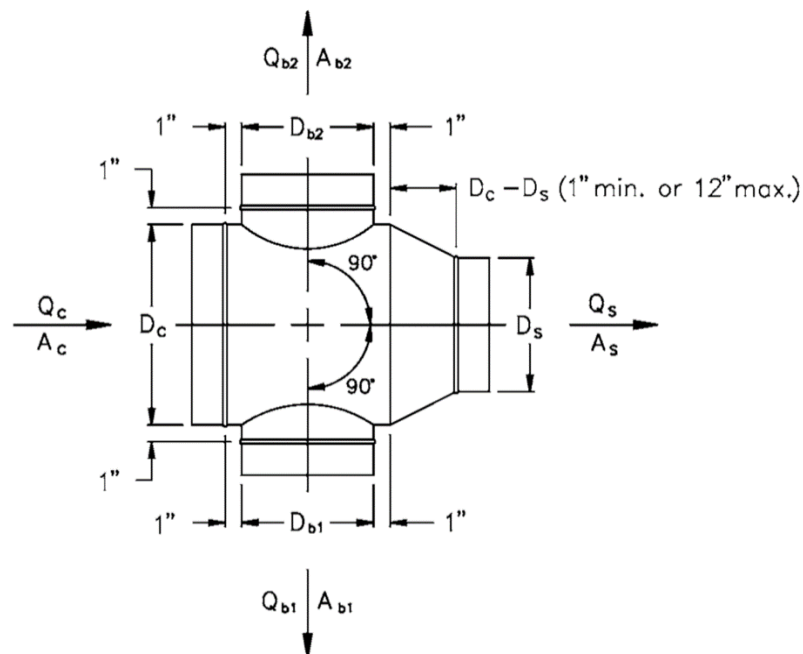

Figure 2.1: Diagram of a double tee section or a cross and its requirements to obtain its corresponding Z-factor.

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3. METHODOLOGY

In order to simulate a real-life situation in which it could be possible a study and variation of the Z-factor, and therefore, the drop pressures, it is necessary to make an experimentation design of an air distribution system in which we could simulate with different diameters of the ducts and different flow rates. A typical air distribution system consists basically in the following parts: a fan which could provide enough initial pressure to all the zones involved with coils for the start of the system, supply ducts divided in branches and return ducts. An appropriate design would be the one which supplies the proper air flow to each of the zones. In this aspect, the pressure in each of the sections of the ducts could presents many issues as it should be high enough to avoid air infiltration into the ducts but not so high in order to prevent from the leaking of the air flow.

3.1. Equal Friction Method

There are many methods for designing an air distribution system. In our case, the Equal Friction Method would be the wiser. This method is based in criteria acquired by professionals along the years and it is easily implementable if we want to build a simple design. One singularity of this method is that the frictional pressure drop per unit length is set at the beginning, depending on if we want to economize in energy costs (lower frictional loss) or in ductwork costs (higher frictional loss). The equal friction method follows the following steps:

- Specify a layout of the design in question with all the parts involved, and the inputs parameters such as the length, necessary flow rates and number and types of fittings.

- Specify a frictional loss per unit length according to *Figure 11.4b* from *Reference [1]*.

- Determine the diameter for each duct section according to the specification of frictional loss per unit length.

- Determine the pressure drop both due to friction and to fittings for each section in order to obtain the total pressure drop to each zone of the system.

- Specify a maximal total pressure in all the zones. In order to achieve a similar pressure in each of the supplied zones, balancing dampers are often needed to increase the flow resistance at the zones with higher pressures.

3.2. Application

In our study case, the Engineering Equation Solver Programme (EES) has been used to determine every parameter in the design. This programme is very useful when, as in our case, we need to know some parameters that are related each other through different equations by beginning with the input parameters that we require for our system. The equal friction method and its implementation in EES is applied in the three following different and successive steps:

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1. **Pre-Design**: an initial experimentation design is needed in order to obtain the appropriate required diameters for each of the duct sections. In this part, balancing dampers are not needed due to it is a previous step before our real design.

2. **Design**: once we know which are the required diameters for achieving the desired flow rates to each of the zones, we build our air distribution system again now with real diameters from the commercial market. In this part, balancing dampers are needed in order to make our design the most realistic possible.

3. **Simulation**: in the last part is time to optimize our design with the same diameters. Flow rates values are not fixed but still the relations between the different flows that pass through each section are considered. Now is time to gradually decrease the static pressure in the end branches of our system (as long as the value approaches to zero our system will be more optimal) in order to obtain new drop pressures with the same Z-factor used for the calculations in the previous step but with different flow rates.

In *Table 3.1* is summarized, in each of the three steps, which are the parameters that come as fixed before the calculation (inputs) and which are the parameters that are obtained and are relevant for further analysis (outputs):

Step	Inputs	Outputs
1. **Pre-Design**	Flow Rate, Frictional Loss per Length	Diameter
2. **Design**	Diameter, Flow Rate	Frictional Loss per Length, Pressure (Total and Static)
3. **Simulation**	Diameter, Static Pressure	Flow Rate, Flow Velocity, Total Pressure,

Table 3.1

Once the simulation is over, it is time to evaluate the feasibility of our system in terms of duct sizes and pressures. Also, as the aim of our project, the study of the drop pressures affected by the Z-factor comes quite relevant as we can analyse them in the particular fittings in our design for different drop pressures and flow rates and see the reliability of the values taken from ASHRAE for the previous calculations.

Figure 2.1 showed the diverging cross which will be applied in our system of study in the sections where the main duct divides in different branches to supply each zone. By obtaining Q_{b1}/Q_c and Q_s/Q_c (The flow rate entering a branch and the flow rate going through the straight duct divided by the flow rate entering the tee) as well as A_s/A_c and A_{b1}/A_c (Cross sectional area of the output and branch section divided by the entering section area) we can determine the Z-factor value for our particular tee. We can find the parametrical tables from ASHRAE in the appendix.

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4. EXPERIMENTATION DESIGN

Figure 4.1 shows the layout of the design applied for our system of study:
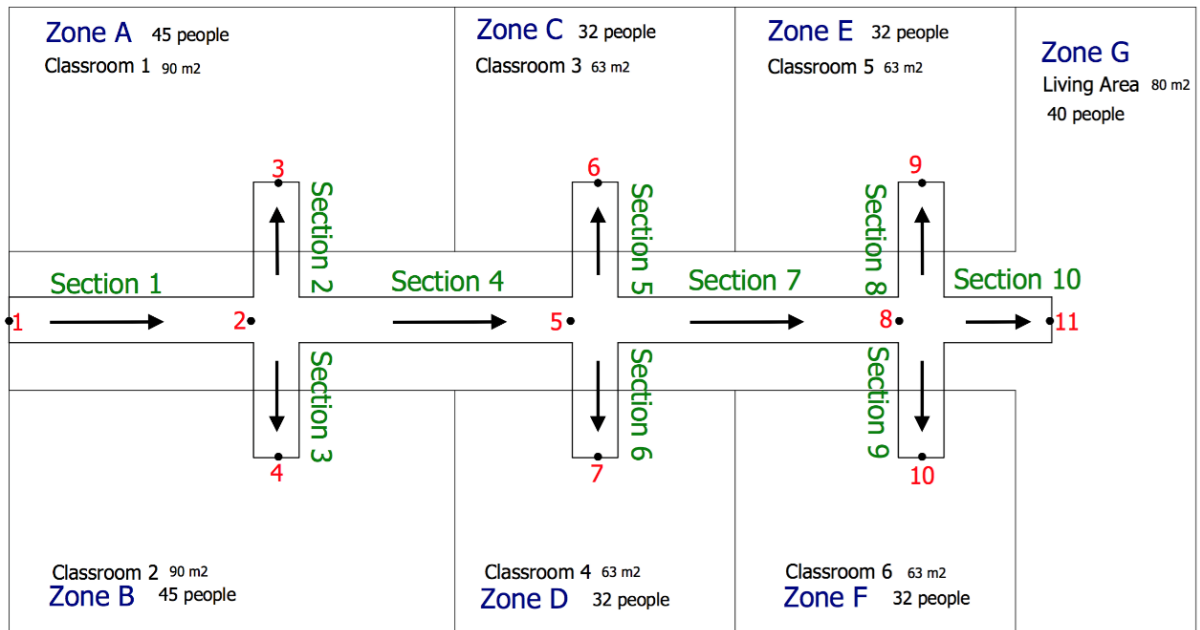
Figure 4.1

Our design consists in an only floor air distribution system expected to be applied in an academic or an office building. Points of study are marked in red, as well as each section of study is determined by the two points in between it. One main duct is supplied by a fan located in the upper part of the building. This makes the supplying duct having a bend just before the first point and thus, a fitting pressure drop is induced in the first section. After the fan, the main duct is divided three times in three different cross-sections with opposite and equal diameters branches to supply each of the different zones with the appropriate air flows. Finally, the last section supplies the remaining air flow to a living area.

Onwards the process of calculations with the EES programme is extensively explained as it follows the three-step methodology that will finally provide us a base for the study of the Z-factors in our actual design. The specific commandos used in EES can be found in the appendix.

4.1. PRE-DESIGN

Firstly, the primitive values of the diameter of the ducts in each section are required for further calculations. The diameter which is going to be taken is the nominal diameter of the duct. Thus, it is measured from between the external diameter and the internal diameter.

Some parameters, such as the atmospheric pressure (101.3 kPa), the viscosity of the air at 20 ºC, the roughness (Ɛ) for medium smooth (value = 0.09 mm, see *Table 11.3* from *Reference [1]*) walls and the density (air = 1.2 kg/m3) have to be specified at the beginning. Also, the specification for of a unit frictional loss of 1 Pa/m for each duct section is inside the recommended range given by ASHRAE for the types of ducts in use (*Figure 11.4b* from *Reference [1]*), but near the lower limits so it will result in a design with low fan energy. Following, the pipe lengths of each duct section. The length is measured

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in the major duct between the first intersection and the second intersection of each section; and in the duct branches between the intersection whom they belong and the end of the branch.

Flow Rates in each duct section

For the flow rates in each section we need a pre-calculation before introducing them into the pre-design. Assuming that our design would be implemented in an educational facility which each zone of the duct branches would be a classroom we have the following data according to ASHRAE, in concept of Ventilation for Acceptable Indoor Air Quality. For a design occupancy of 50 people each 100 m^2, we have an established minimum airflow rate per person of 7.5 L/s (*Table 7.3* from *Reference [1]*).

According to the surface in each room that means that the perfect design occupancy in Zones A and B are 45 people; in Zones D, E and F are 32 people and in Zone G 40 people. This translates into the following minimum flow rates (in m^3/s) for the different rooms/zones, and also corresponding for the end points of the branches.

Zone/End Point	Flow Rate (m^3/s)	Zone/End Point	Flow Rate (m^3/s)
A / $\dot{V}_2$	0.34	E / $\dot{V}_8$	0.24
B / $\dot{V}_3$	0.34	F / $\dot{V}_9$	0.24
C / $\dot{V}_5$	0.24	G / $\dot{V}_{10}$	0.30
D / $\dot{V}_6$	0.24		

Given the minimum flow rate in the last zone, we can calculate the flow rates in each duct section by making sure that the flow rate in the Section 1 is enough to supply all the zones (*Equations 4.1, 4.2 & 4.3*).

$$\dot{V}_1 = \dot{V}_2 + \dot{V}_3 + \dot{V}_4 \quad (4.1) \qquad \dot{V}_4 = \dot{V}_5 + \dot{V}_6 + \dot{V}_7 \quad (4.2) \qquad \dot{V}_7 = \dot{V}_8 + \dot{V}_9 + \dot{V}_{10} \quad (4.3)$$

Fitting Pressure Drops

The values of the loss coefficients for the system in use are the following:

Fitting	Z_L
Entrance	0.05
Bend	0.1
Diffuser	0.1

This translates into the followings sums of fitting pressure drop coefficients for each section (*Equations 4.4, 4.5, 4.6 & 4.7*):

$$SumZ_1 = 3*Z_{bend} + Z_{ent} \tag{4.4}$$
$$SumZ_9 = SumZ_8 = SumZ_6 = SumZ_5 = SumZ_3 = SumZ_2 = 3*Z_{bend} + Z_{diffuser} \tag{4.5}$$
$$SumZ_4 = SumZ_7 = 0 \tag{4.6}$$
$$SumZ_{10} = 9*Z_{bend} + Z_{diffuser} \tag{4.7}$$

With all this previous information we can build the following table (*Table 4.1*):

Section	Location	Length (m)	Flow Rate (m³/s)	Sum of Z_L
1	1 to 2	6	2	0.35
2	2 to 3	4	0.35	0.4
3	2 to 4	4	0.35	0.4
4	2 to 5	7	1.3	0
5	5 to 6	4	0.25	0.4
6	5 to 7	4	0.25	0.4
7	5 to 8	7.5	0.8	0
8	8 to 9	4	0.25	0.4
9	8 to 10	4	0.25	0.4
10	8 to 11	5	0.3	1

Table 4.1

Iteration

Before calculating the pressure drops we need to stablish an iteration in which some magnitudes such as the friction loss depends on the diameter of the duct and the flow velocity as well as the flow velocity depends on the diameter. Mass rate, area, duct diameter, Reynolds number and friction factor are needed. All these magnitudes are referred to each section (*i = 1, … , 10*). For the last mentioned we use the expression for rough walls in a duct with a supposed Reynold number lower than 10^6 (*Equation 4.8*, see *Table 3.2* from *Reference [1]*).

$$f_i^{-5} = 1.14 + 2*\text{Log}(DE_i) - 2*\text{Log}\left(1 + \frac{9.3}{\frac{Re*f_i^{-5}}{DE_i}}\right) \tag{4.8}$$

With $DE_i = \frac{\text{Diameter}_i}{\text{Roucghness}_i}$.

Then we determine the dynamic pressure in each section (*Equation 4.9*) and the pressure drop due to friction (*Equation 4.10*) and as well as we define the frictional loss per length specified before (*Equation 4.11*):

$$VP_i = \rho * \frac{V_i^2}{2} \tag{4.9}$$

Being $V = \frac{V_1}{A_i}$

$$DP_{\text{friction},i} = \left(f_i * \frac{L_i}{DH_i}\right) * VP_i \tag{4.10}$$

$$DPL_i = \frac{DP_i}{L_i} \tag{4.11}$$

With this and after the iterative process we can already display the resulting diameters, flow velocities and total friction pressure drop (*Table 4.2*) as well as the other magnitudes (*Table 4.3*).

Section	Specified Friction Loss (Pa/m)	Required Diameter (m)	Flow Velocity (m/s)	Friction Pressure Drop (Pa)
1	1	0.5736	7.74	6
2	1	0.2974	5.04	4
3	1	0.2974	5.04	4
4	1	0.4874	6.968	7
5	1	0.2621	4.634	4
6	1	0.2621	4.634	4
7	1	0.4058	6.185	7.5
8	1	0.2621	4.634	4
9	1	0.2621	4.634	4
10	1	0.2806	4.85	5

Table 4.2

We can notice that the resulted velocities for each section are close to the recommended design velocities for air distribution systems (*Table 4.4* or *Table 11.4* from *Reference [1]*).

Pressure drops

The following calculations are only necessary for the design step, but will be explained here for a better a better reading.

The representative loss coefficients for a diverging cross are given in ASHRAE (*Chapter 1, SD5-24*, *Reference [2]*), as in the example in *Table 11.2* from *Reference [1]*:

Point before the cross	$Z_{cross,straight}$	$Z_{cross,branch}$
2	0.13	2.4
5	0.14	2.4
8	0.14	1.94

The pressure drop in each section due to the fittings are calculated by adding the sum of the loss coefficients for each section and the representative loss coefficients to the expression of the dynamic pressure, as for example with *Equation 4.12*:

$$DP_{fittings,i} = SumZ_i * VP_i + Z_i * \rho * \frac{V_{previous\ section\ at\ the\ cross}^2}{2} \tag{4.12}$$

We set the total pressure at the last point (pt_{11}, in contact with the last zone) in 70 Pa atmospheric to make sure the pressure in each zone is above a reasonable value. Thus, the total pressure for each location, adding the friction pressure drop to the pressure drop due to the fittings is the following (*Equation 4.13*):

$$Pt_{previous\ section\ at\ the\ cross} - Pt_{i+1} = DP_{friction,i} + DP_{fittings,i} \tag{4.13}$$

So then, the total pressure drop in each duct section both due to friction and fittings is (*Equation 4.14*):

$$DPt_i = Pt_{previous\ section\ at\ the\ cross} + Pt_{i+1} \tag{4.14}$$

With the consequent displayed results in *Table 4.5*

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Likewise, the dynamic pressure of each section is subtracted from the total pressure at each point which corresponds with that section to obtain the static pressure without balancing dampers (*Equation 4.15*, results in *Table 4.6*):

$$Pstat_i = Pt_i\text{-}VP_{\text{section where the point is located}} \qquad (4.15)$$

Point	1	2	3	4	5	6	7	8	9	10	11
Static Pressure (Pa)	98.21	79.63	3.972	3.972	74.78	11.96	11.96	69.68	25.77	25.77	55.89

Table 4.6

As well as the pressure drop to each zone; being *i = A, … , G* (*Equation 4.16*, results in *Table 4.7*):

$$DPZ_i = Pt_1\text{-}Pt_{\text{point in contact with the zone}} \qquad (4.16)$$

Zone	A	B	C	D	E	F	G
Pressure Drop to Zone (Pa)	114.9	114.9	109.3	109.3	95.51	95.51	64.16

Table 4.7

4.2. DESIGN

Diameters in each duct section

Once we have obtained the primitive values for the duct diameters we require the real values from a commercial catalogue of nominal diameter in a duct. These are the common circular duct sizes used in air ventilation systems (*Reference* [7]):

Nominal Diameter (mm)													
63	80	100	125	160	200	250	315	400	500	630	800	10000	1250

Thus, the inputs in our design will be the new duct diameters accordingly with the commercial sizes and the flow rates that we stated in the pre-design (*Table 4.8*):

Section	Required Diameter (m)	Nominal Diameter (m)	Flow Rates (m³/s)
1	0.5736	0.630	2
2	0.2974	0.315	0.35
3	0.2974	0.315	0.35
4	0.4874	0.500	1.3
5	0.2621	0.315	0.25
6	0.2621	0.315	0.25
7	0.4058	0.500	0.8
8	0.2621	0.315	0.25
9	0.2621	0.315	0.25
10	0.2806	0.315	0.3

Table 4.8

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Iteration

With the new duct sizes, the commandos for the calculation of the parameters in the iterative process are the same as explained before but, in this case, the output is the frictional loss per meter, which now has to be calculated to translate our design into a real situation. Also, the obtaining the drop pressures in each section and the dynamic pressures is the same. As we are working with the same design the length of the sections and the sum of the loss coefficients remain constant, but the friction pressure drops changed due to the new frictional loss. With all this the results are displayed in the *Tables 4.9 & 4.10*.

Section	1	2	3	4	5	6	7	8	9	10
Flow Velocity (m/s)	6.42	4.49	4.49	6.62	3.21	3.21	4.07	3.21	3.21	3.85
Frictional Loss (Pa/m)	0.6289	0.7536	0.7536	0.8815	0.4069	0.4069	0.3586	0.4069	0.4069	0.5678
Friction Pressure Drop (Pa)	3.77	3.01	3.01	6.17	1.63	1.63	2.69	1.63	1.63	2.84

Table 4.9

As we can notice, the resulted flow velocities in this case are slightly lower than in the pre-design due to an upper duct size, so that they fit into the recommended ranges for air distribution systems (*Table 11.4* from *Reference* [1]).

Pressure Drops

In this case we have different representative loss coefficients for each cross in our system than before due to that the duct sizes have changed. So that, the loss coefficients in use are:

Point before the cross	$Z_{cross,straight}$	$Z_{cross,branch}$
2	0.14	2.4
5	0.17	4.1
8	0.13	1.94

With them the results in pressure drops due to fittings along with the total pressure drop in each section and to each zone are displayed in *Table 4.11 & 4.12*. We can also state the following equality in order to make sure the dynamic pressure of each point matches with the static pressure in it and avoid confusing between dynamic pressure in sections and in points (*Equation 4.17*):

$$\text{Pdynamic}_{\text{point}} = \text{VP}_{\text{section where the point is located}} \tag{4.17}$$

This time we set the total pressure at the last point in 100 Pa above atmospheric in order to ensure there are no negative total pressures in the other ends of the branches. Furthermore, this will allow us to keep the static pressure the minimal possible in each end of branch, though it could not be the optimal design. An optimal one means that the air flow supplied to each room is the best suitable, as it can provide ventilation in the desired quantity (as long as the static pressure at the end of the branch approaches to 0 would result in a more suitable flow).

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Still, the calculation of the static pressure is the same as in the pre-design, and its results are displayed in *Table 4.13*:

| Point	1	2	3	4	5	6	7	8	9	10	11
Static Pressure (Pa)	117.50	105.10	50.58	50.58	93.88	2.08	2.08	103.10	83.43	83.43	91.11

Table 4.13

In order to ensure a total pressure lower than 100 Pa in all the points at the end of the branches it would be necessary to apply balancing dampers, though this is not the case. Anyway, the pressure drops across balancing dampers to zones that would be at higher pressure than the last zone would be calculated as in *Equation 4.18*:

$$\text{DPbal}_{\text{zone}} = \text{Pt}_{\text{point in contact with the zone}} - \text{Pt}_{11} \qquad (4.18)$$

The resulted data is displayed in the following graphs:
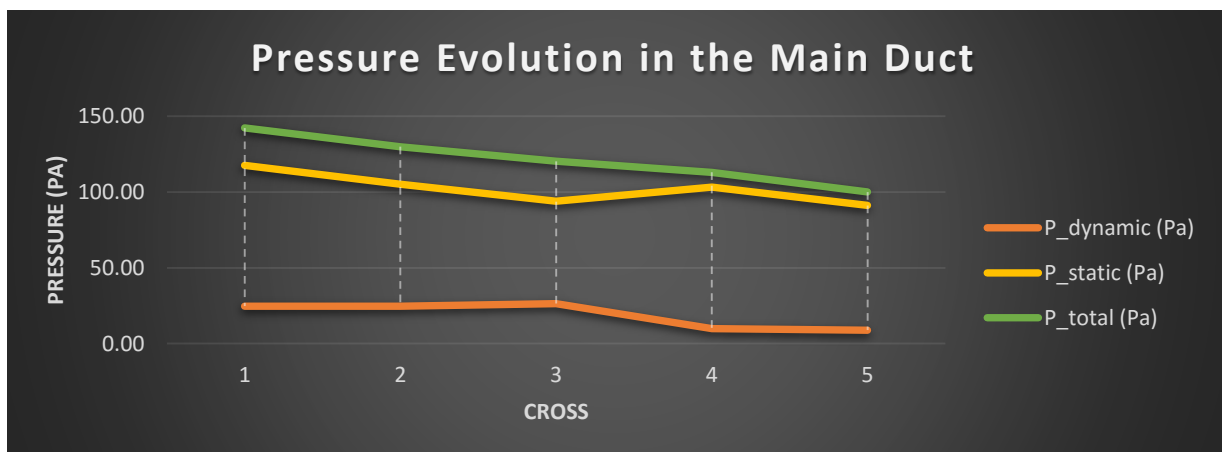
Graph 4.1
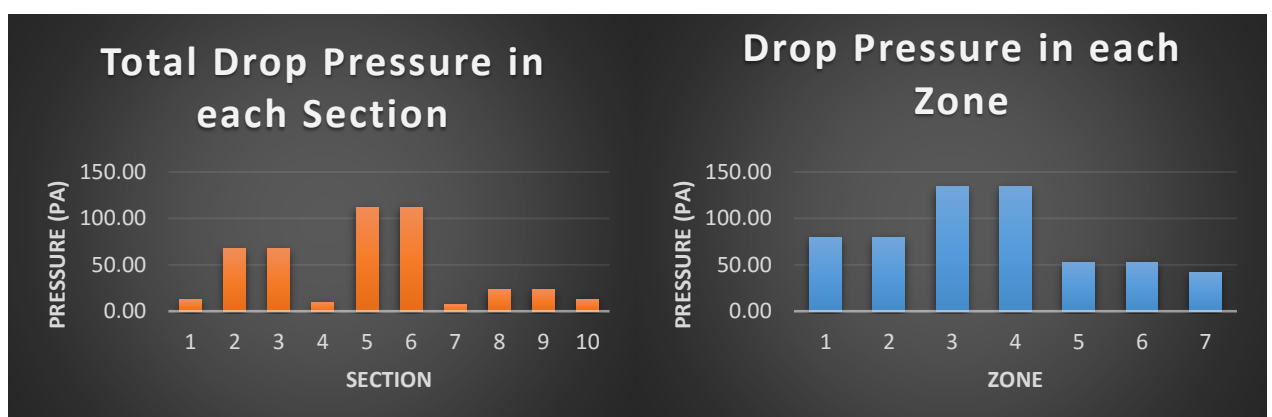
Graphs 4.2 & 4.3

- It is remarkable that the pressure drops in the second cross, referring to the section 5 and 6) is quite higher in comparison with the rest of the crosses (*Graph 4.2*) due to the Z factor values given by ASHRAE. Also, there is a static pressure regain after the second cross.

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4.3. SIMULATION

The results in static pressure at the end of the branches require to be optimize. As our objective is to approach them the most possible to zero, now is time to simulate with the same parameters that we had in our design several times until an optical values of static pressure are reached. This time flow rates are left as opened variables as well as the static pressure at the points in the main duct (1, 2, 5 and 8). The points at the end of the branches and the last one (3, 4, 6, 7, 9, 10 and 11) are firstly set with the values obtained in the design step and are being decreased in successive simulations. In every simulation we take the flow rate given as a result by the previous simulation and new values of Z-factor are taken from the tables in ASHRAE, to use in the next simulation. The values of Z-factor both for the straight section and for the branch section in the diverging cross are the same due to the high tolerance of these values, until we reach a point where the variation of flow rate is high enough to make a difference.

Hereby are displayed the optimal values of static pressure (*Table 4.14*):

Point	1	2	3	4	5	6	7	8	9	10	11
Static Pressure (Pa)	100.8	87.06	20	20	80.04	0	0	88.52	55	55	76.11

Table 4.14

Also, some inputs of diameter sizes have to be changed to reach the minimum flow rates in Areas C and D. The changed diameters in the branches of the second cross along with the corresponding new flow rates are shown in *Table 4.15*:

Section	1	2	3	4	5	6	7	8	9	10
Diameter (m)	0.630	0.315	0.315	0.500	0.400	0.400	0.500	0.250	0.250	0.315
Flow Rate (m³/s)	**2.10**	**0.422**	**0.422**	**1.26**	**0.240**	**0.240**	**0.779**	**0.240**	**0.240**	**0.3**

Table 4.15

When this optimal design is reached new values of Z-factor are found according to the new relations Q_b/Q_c and A_b/A_c in the cross sections (*Table 4.16*):

	$Z_{cross1,branch}$	$Z_{cross2,branch}$	$Z_{cross3,branch}$	$Z_{cross1,straight}$	$Z_{cross1, straight}$	$Z_{cross1, straight}$
Design	2.4	4.1	1.94	0.14	0.17	0.13
Pre-Simulation (new flow rates)	2.4	15.89	1.94	0.13	0.12	0.13
Simulation (new flow rates and diameters)	2.4	8.99	1.2	0.13	0.17	0.13

Table 4.16

This is why the optimal design has been reached as we have found certain values of Z-factors, as for example 15,89 or 8,99 in our cross-section that would change significantly the air flow in the next simulation step (it would be too low for reaching the minimum for its corresponding zone).

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5. STUDY OF Z-FACTORS IN T-SECTIONS

The study of the Z-factor in different tees cannot be obtained by means of calculations on its own. To obtain values, it has to be done in an empirical way, therefore a practical experiment is required. A simple experiment can be carried out to determine different values for the Z-factor of each fitting. An example of this is illustrated in Figure 5.1, with all the instrumentation labelled.
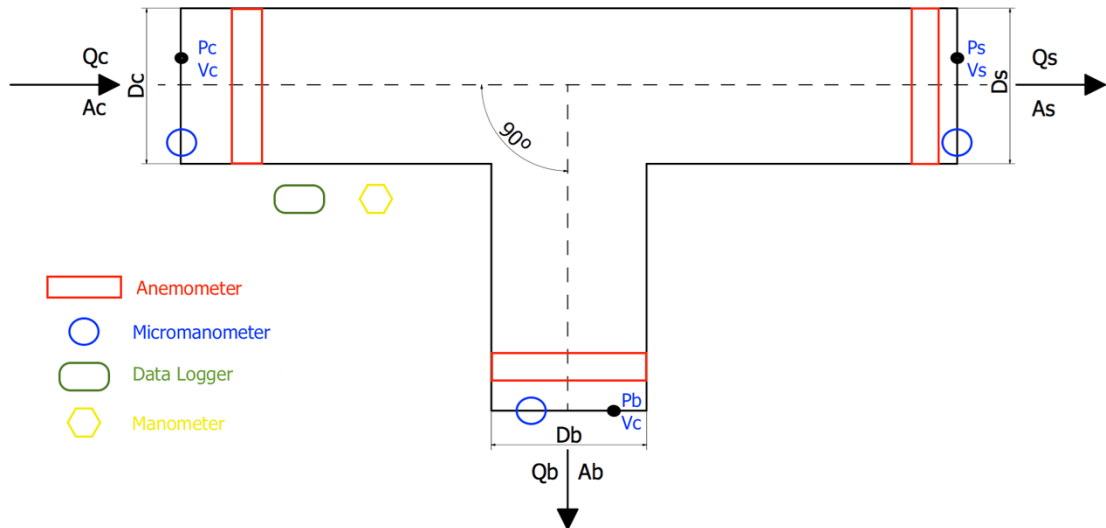

Figure 5.1: Diagram of a simple 90º tee section.

The data that has to be obtained to calculate values for Z-factors is based on *Equation 5.1*:

$$\Delta P_f = Z \cdot \frac{\rho \cdot v^2}{2} \tag{5.1}$$

Thus, the parameters such us the pressure drop due to fittings, the density and the flow velocity are required to be obtained empirically. The data that could allow us to obtain all the variables is shown in *Table 5.1*:

Variables Required:	Units:	Instrumentation:	Brand and Model:	Precision Error:
Diameter of all the sections	(m)	Measuring tape	Stanley 33-725	±0.005 m
Pressure entering the tee	(Pa)	Micromanometer	Fluke 922	±1% + 1Pa
Pressure exiting the tee	(Pa)	Micromanometer	Fluke 922	±1% + 1Pa
Pressure at the branch	(Pa)	Micromanometer	Fluke 922	±1% + 1Pa
Velocity of the fluid in the different sections	(m/s)	Anemometer	Pyle PMA90	±3% + 0.2m/s
Air Pressure	(kPa)	Manometer	UEI-EM201	±0.05%
Fluid Temperature	(ºC)	Data Logger	Testo 174 T	±0.5 ºC

Table 5.1: Variables required to calculate the Z-factor.

First of all, the density of the fluid has to be obtained. The density depends on the temperature and pressure of the fluid and will be kept as constant throughout the entire experiment. Using EES we can determine the density of the air flowing by knowing the air pressure and temperature.

Temperature and pressure will be measured outside the system using a manometer and a data logger respectively to obtain the air conditions.

By measuring all the ducts internal diameters, the cross sectional area of each section can be obtained. A fan is in charge of providing the air into the system.

Micromanometers will record pressure values of the different sections, at the point just before the end of the duct. In a similar way, Anemometers located in the same position as the micrometres will be in charge of measuring the air's velocity.

Once all the variables are obtained, the Z-factor can be calculated with *Equation 5.1*. The pressure drops in each section can be determined by subtracting the final pressure to the initial pressure. With the calculated drop pressure, the density and the velocity of the section we can finally determine a value for the Z-factor.

It has to be said that this value will carry an error. Each instrument possess an error which can be due to its calibration or its precision. The error for the obtained Z-factors will be induced by the instrumentation errors and it can be obtained by the following formulas (*Equations 5.2 & 5.3*):

$$\text{Absolute error: } \varepsilon_a = \overline{X}\text{-}X_i \qquad (5.2)$$

$\overline{X}$: Mean value of all the recordings
X_i: Obtained value from the measurement

$$\text{Relative error: } \varepsilon_r = \frac{\varepsilon_a}{\overline{X}}.\,100\% \qquad (5.3)$$

Once some experimental values for Z-factors have been obtained, they can be compared to other sources, such as ASHRAE. By knowing the areas and flow rates for the tees, Z-factors can be obtained in ASHRAE tables and be compared to the experimental values.

6. CONCLUSIONS

Along this project we have followed the designing process of a HVAC system in a particular area that can be applied for an office building. Because of it is a very specific discipline in the field of the Fluid Mechanics, a thorough research has previously been necessary. Sources such as books, scientific articles and webpages have been the base of research for this case study. Between the quantity of information that is available in concept of designing a system with tee sections like ours, there is not much information on Z-factors and its causes. This drew our attention considerably so it had an immediate reaction upon us, which lead us into an interest for us to focus on its study.

In order to carry out the design, three steps have been required: Pre-Design, Design and Simulation. Different data has been obtained from each of them and different data has been fixed or left as variable.

In the Pre-design step we have obtained the diameter of the ducts by fixing the flow rates in each zone and the frictional loss per unit length. Our main focus in this step is to give dimensions to the ducts in our system. Once we obtain theoretical values for the dimeters, we choose normalised sizes and size the system.

Secondly, in the Design step we work with the real size of ducts and keep the value of the flow rates. The frictional loss per unit length is nt fixed any more in order to calculate real values for the drop pressures.

Finally, in the Simulation step we have refined our system by altering the static pressures in the branches without exceeding the minimum required flow rates.

As it has been proved in the analysis, the value of the Z-factor in our tee section of study is determined by the relationships between the flow rate in the branch and the main duct as well as the relationship between area of the branch and the main duct. With this said, it is of extreme importance to avoid having a low quotient of flow rates as well as high quotients of areas because these will lead into high Z-factor values that will cause havoc in the drop pressures due to the tee section. Therefore it is advisable to keep the relationships of flow rates and areas around the central zone of the table of Z-factor values (around 0.3~0.7 for flow rates and around 0.1~0.5 for the areas).

These values given by ASHRAE affect considerably to the design of the system that they alter dramatically the pressures in each point. This is to say that, for intermediate values from between two consecutive quotients of flow rates or areas, they do not correspond to a proportional variation of the quotients mentioned but to a randomly changing values from small to very large.

Finally, it has been proposed an experiment to obtain empirically values of Z-factor (with its corresponding errors to be considered) in a certain tee section in order to compare the obtained values with the ones given by ASHRAE and prove the reliability of these tables of Z-factors.

We can conclude that, trusting the given values for the Z-factors, we are able to develop an experiment that later allows us to test out these values and confirm the reliability of them.

7. BIBLIOGRAPHY

- [1] Principles of Heating, Ventilation and Air Conditioning in Buildings, 2013 Ed. Wiley. John W. Mitchell & James E. Braun

- [2] 2009 ASHRAE Handbook - Fundamentals

- [3] Heating, Ventilating and Air Conditioning: Analysis and Design, Ed. Wiley. McQuiston & Parker

- [4] Vassernan, Kazavchinskii, and Rabinovich, "Thermophysical Properties of Air and Air Components;' Moscow, Nauka, 1966, and NBS-NSF Trans. TT 70-50095, 1971

- [5] Vassernan and Rabinovich, "Thermophysical Properties of Liquid Air and Its Component, "Moscow, 1968, and NBS-NSF Trans. 69-55092, 1970

- [6] http://validyne.com/blog/application-note-basics-of-air-velocity-pressure-and-flow/

- [7] http://www.engineeringtoolbox.com/surface-roughness-ventilation-ducts-d_209.html

- [8] http://www.uvm.edu/~pdodds/teaching/courses/2009-08UVM-300/docs/others/2007/vasava2007a.pdf

- [9] http://www.monografias.com/trabajos-pdf3/ecuacion-explicita-calculo-friccion/ecuacion-explicita-calculo-friccion.pdf