MASTER

Possible energy savings at Aircraft Fuel Supply

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Possible Energy Savings
at Aircraft Fuel Supply

This report covers parts of my graduating project at Shell. This graduating project is part of my study Electrical Engineering at the Eindhoven University of Technology.

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Preview

From October 15, 1996 until June 12, 1997 I have performed my graduating project at Shell, as part of my study Electrical Engineering at the Eindhoven University of Technology. I have learned a lot from this project and enjoyed it very much. I believe it is very important for a student to get some practical experience in a company. This report describes most of the work I have performed during this period.

I would like to thank Shell and AFS for giving me the opportunity to perform my graduating project here. In particular I would like to thank ing. J. de Wit and ir. C. van Ouwerkerk for making everything possible. Finally I would like to thank Prof.dr.ir. P.P.J. van den Bosch at the Eindhoven University of Technology for accepting the difficult task to be external supervisor.

Summary

This report deals with the possibility of future energy savings at Aircraft Fuel Supply (AFS) at Schiphol Airport. When an amount of kerosene is demanded at the piers at Schiphol, AFS takes care of it.

Reviewing energy bills is always the first step in making a good energy evaluation. From this evaluation it is concluded that AFS has paid too much in 1996 and that the pumps are the main energy users (86%). So the pumps at AFS were evaluated.

By keeping in mind the main policy of AFS, delivering a certain flow at any time at Schiphol, it turned out that the best way of saving energy is reducing the produced head of the pumps. The best way to do this, is changing the speed of the pumps. This can be accomplished by Variable Speed Drives. The best solution with respect to rangeability, speed, control accuracy, and maintenance is an implementation of Variable Frequency Drives (VFD's).

By calculating the savings and investment costs and keeping in mind taxes, government subsidies, additional advantages and future perspectives, it is recommended that AFS should install two Variable Speed Pumps by using VFD's.

List of Abbreviations

AC	Alternating Current
AFS	Aircraft Fuel Supply
ASP	Amsterdam-Schiphol Pipeline
A&E	Architects and Engineering
BEF	Best Efficiency Flow
CSI	Current Source Input
DC	Direct Current
DPO	Defence Pipeline Organisation
DRR	Discounted cash flow Rate of Return
ECC	Eddy-Current Coupling
EIA	Energie Investerings Aftrek
FFT	Fast Fourier Transform
FSP	Fixed Speed Pump
CCF	Cumulative Cash Flow
CF	Cash Flow
GS	Gross Savings
HF	Harmonic Factor
IT	Income Taxes
ITHD	Current Total Harmonic Distortion
NEC	National Electric Code
NPSH	Net Positive Suction Head
PCC	Point of Common Coupling
PWM	Pulse Width Modulated
SCR	Semiconductor-Controlled Rectifiers
STP	Simple Time to Payback
TD	Tax Depreciation
TI	Taxable Income
TDH	Total Dynamic Head
THD	Total Harmonic Distortion
VFD	Variable Frequency Drive
VSD	Variable Speed Drive
VSM	Variable Speed Motor
VSP	Variable Speed Pump
VTHD	Voltage Total Harmonic Distortion
VVI	Variable-Voltage Input

List of Symbols

a	Attenuation factor	[-]
$\mathbf{A_N}$	Notch area	[secV]
C	Capacitance	[F]
$\mathbf{\cos\theta}$	Power factor	[-]
f	Frequency	[Hz]
$\mathbf{f_R}$	Resonant frequency	[Hz]
g	Graviational acceleration	[m/sec^2]
H	Head	[m]
$\mathbf{H_{dyn}}$	Dynamic head	[m]
$\mathbf{H_S}$	Pumping system characteristic	[m]
$\mathbf{H_{st}}$	Static head	[m]
$\mathbf{H_v}$	Head loss	[m]
h	Harmonic order	[-]
I	Current	[A]
$\mathbf{I_b}$	Base current	[A]
$\mathbf{I_N}$	Nominal current	[A]
$\mathbf{I_{SC}}$	Short circuit current	[A]
$\mathbf{I_\Delta}$	Start-up current, deltaconnection	[A]
$\mathbf{I_Y}$	Start-up current, starconnection	[A]
k	pipe internal surface roughness	[m]
L	Inductance	[H]
n	Operating speed	[rpm]
$\mathbf{n_S}$	Synchronous speed	[rpm]
$\mathbf{N_S}$	Specific speed	[-]
P	Power	[kW]
$\mathbf{P_1}$	Active power absorbed	[kW]
$\mathbf{P_b}$	Frictional losses in the balancing device	[kW]
$\mathbf{P_d}$	Disc friction losses	[kW]
$\mathbf{P_{co}}$	Connection losses	[kW]
$\mathbf{P_m}$	Mechanical losses	[kW]
$\mathbf{P_Q}$	Effective power transmitted to the liquid	[kW]
$\mathbf{P_S}$	Reactive power	[kW]

Q	Flow rate	[m^3/h]
Q_B	Balancing flow rate	[m^3/h]
$Q_{centrum}$	Flow demanded at the centum	[m^3/h]
Q_D	Flow demanded at the D-pier	[m^3/h]
Q_E	Flow demanded at the E-pier	[m^3/h]
Q_F	Flow demanded at the F-pier	[m^3/h]
Q_G	Flow demanded at the G-pier	[m^3/h]
Q_L	Leakage flow rate	[m^3/h]
R	Resistance	[Ω]
Re	Reynolds number	[-]
s	Slip	[%]
T	Torque	[Nm]
T_A	Starting torque at standstill	[Nm]
T_K	Pull-out torque	[Nm]
T_N	Nominal torque	[Nm]
T_S	Pull-up torque	[Nm]
T_Δ	Motortorque, deltaconnection	[Nm]
T_Y	Motortorque, starconnection	[Nm]
U_N	Nominal voltage	[V]
u	Investment tax rate	[%]
v	Velocity	[m/s]
z	Corporate tax rate	[%]
Z_f	Impedance of the filter	[Ω]
α	Detuning factor	[-]
η	Efficiency	[%]
η_{co}	Connection efficiency	[%]
η_h	Hydraulic efficiency	[%]
η_m	Mechanical efficiency	[%]
η_{mo}	Motor efficiency	[%]
η_v	Volumetric efficiency	[%]
λ	Coefficient of friction	[-]
ν	Kinematic velocity	[mm^2/s]
ρ	Density of the liquid	[kg/m^3]
ω	Angular velocity	[rad/s]

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Chapter 1

Introduction

Petroleum and chemical plants of today are effectively cutting energy losses in their plants thermally, electrically, and mechanically in their process equipment. In rotating process equipment such as pumps, fans, compressors, and blowers, much mechanical or fluid energy is being dissipated or thrown away by throttling valves, dampers, and adjustable guide vanes. Adjustable speed is an increasingly used feature in the search to find energy losses and reduce them.

As electrical power costs continue to climb more and more attention is being paid to reducing electrical losses, principally by turning to cogeneration, more efficient distribution equipment, high efficiency motors, and adjustable speed drives.

It is noticed that for every kilowatt of loss saved in the electrical utilization equipment or process drives means that 6kW of fuel energy has been saved on the front end of the system [Hickok, 1983]. This point brings into sharper focus the need to find and save losses in process drives. Process drive losses are not always easy to find. They are pretty much silent and concealed in mechanical, electrical, fluid, and chemical characteristics of a process and its equipment. They must be dug for in characteristic curves, fluid mechanics, process theory, work equations, and electrical circuits. Because a number of disciplines are involved, the trend today is to put together an energy saving team.

A combination of mechanical, electrical, chemical, process, and instrumentation engineers are assembled to proceed through the plant on a systematic energy audit.

An engineering-economic evaluation can incorporate a variety of analyses. Reviewing energy bills is always the first step. As a first step, the engineer obtains energy bills paid by the utility and determines the fraction used by each user.

Such a review of energy bills detects whether the unit cost changes with the time or year or whether a particular distribution station has higher unit costs than others. This analysis may even identify billing errors on the part of the electric utility. Finally, the review should give the utility an idea of the magnitude of savings possible using more sophisticated techniques to reduce energy costs. This review of the energy bills will be discussed in the next chapter.

Aircraft Fuel Supply is a pumping station which pumps the JET A-1 fuel to Schiphol Centre. This is done by twelve centrifugal pumps driven by electric motors. If one wants to determine potential savings, the following information is required:

- **The pump performance charts**. This will be dealt with in chapter three. The electric motors which drive these pumps are discussed in chapter four.
- **The system curves** (Chapter six).
- **The duty cycle of the system**. This is the key element in determining potential savings and is frequently highly variable. A common approach to obtaining a duty cycle estimate is estimating the amount of time spent at several pressure or flow points depending on the system (At AFS this is the flow).
- **The control methods and efficiency curves**. The control of pumps is discussed in the chapters five and seven.

Before beginning a detailed system analysis, it is necessary to outline the system configuration and the methods of control. This would require that three key elements should be considered:

- **the control loops**: How will the system operate and how the system is to be analysed. There may be many control loops on a given system with various effects, but in most cases the system can be reduced to a fairly simple block diagram.
- **the control methods and drive components**, (chapter five, eight and nine) which include:
    - throttling valves;
    - bypass loops;
    - ac motors;
    - slip-type drives;
    - variable frequency drives;
- **the system configuration**:

- Defining the type of electric motor that is used, whether standard efficient of energy-efficient. The efficiency characteristics change when a motor is operated on adjustable-frequency power and, in fact, are different for each type of adjustable-frequency controller. This should be taken into account when analysing a system.

- Defining the type of drive. Common types of adjustable-speed drives include slip-type drives (include the eddy-current couplings and fluid couplings, so the efficiency will decrease linearly with the speed) and adjustable-frequency controllers such as:
  1. Variable-voltage input (VVI);
  2. Current source input (CSI);
  3. Pulsewidth modulated (PWM).

  The efficiencies of each type are different.
- With respect to drive efficiencies consisting on combined motor and controllers, the controller waveform will affect the motor efficiency due to fixed losses in the controller. Due to this interaction, it is desirable to obtain an overall drive efficiency from the drive vendor consisting of the combined motor and controller efficiency.

Each of the three controllers mentioned before impacts the power quality, power factor (chapter ten) and harmonic content (chapter eleven).

Detailed system analysis

After obtaining all the information mentioned above, the input power at each operating point may be calculated for the purposes of comparing two alternatives. This would be done by starting at the process and calculating the input power required for the system by using the efficiency information for each component.

After calculating the input power at each operating point and by using the duty cycle in a year, the total amount of absorbed energy in a year [kWh] can be calculated:

$$E = P_i * T_i$$

where P [kW]
T [h]

Now the savings between alternatives can be calculated and then be used as the basis of an economic payback (chapter twelve).

Economic analysis

After determining the expected savings between alternative control methods, the savings can be used as the basis for a payback analysis. To do this, however, all required hardware and installation costs must be defined (chapter thirteen).

Future perspectives

It is very important to keep in mind the future perspectives of both AFS and Schiphol (chapter fourteen).

Chapter 2

Situation at Aircraft Fuel Supply

2.1 General

Although the oil companies at Amsterdam Airport Schiphol are competitors when it comes to selling fuel to customers such as airlines, they work closely together on the receipt and handling of JET A-1 (kerosine) aviation fuel at the airport. To this end, they have joined forces in Aircraft Fuel Supply (AFS) B.V.

AFS is responsible for the construction and operation of all fixed fuel supply facilities at Schiphol. The kerosine can be supplied by means of:

1. Defence Pipeline Organization (DPO) pipeline, which connects Shiphol Airport direct to the oil refineries of Rotterdam;
2. Inland tankers. AFS operates three special jetties on the Ringvaart canal to unload the fuel from the barges. The fuel is pumped from here into two receiving tanks:
    - -tank 931, which has a capacity of 1,500m^3;
    - -tank 932, which has a capacity of 2,000m^3.

These two tanks merely serve as a buffer to compensate for irregularities in the speed with which the Jet A-1 fuel is delivered. The fuel is not stored here, but booster pumps transport it direct (8,500 litres a minute) to the RijkI storage depot. The pipeline system of this part (East) is given in Appendix A1.

The RijkI storage depot is divided into two parts. The first part RijkIA is not used anymore. The second part RijkIB consists of four tanks (106 through 108). They have a capacity of 6,800m^3. RijkIB is given in Appendix A2.

The fuel being transported by DPO pipeline is delivered direct to the receiving tanks (201 through 203) at the RijkIIA depot. There it has to settle for 24 hours. Afterwards, three intertank transfer pumps take care of delivery from the receiving tanks to the storage tanks (204 through 206) at the Rijk IIB depot. The tanks 201 through 206 can store 12.500m^3 each. The RijkII depot is given in Appendix A3;

Twelve hydrant pumps transport the Jet A-1 in quantities of some 5,000 litres a minute each from the Rijk depot to Schiphol Centre. On the one hand it is transported to the Octaanplein site, from where large refuellers (road tankers) depart to fuel aircraft parked at the B,C and part of the D pier and on the cargo aprons. After receipt and storage AFS supplies the fuel, but the actual fuelling of aircraft is not one of its tasks.

On the other hand, fuel is transported around the D, E, F, and G piers via an underground hydrant system. Here, AFS supplies to the 103 hydrant pits in the apron. To secure that there will be always delivery via the hydrant, the hydrantlines around the passenger piers are not only connected to the main feederline, but also interconnected to each other by means of back-up lines. Dispensers are used for fuelling from these locations.

AFS's throughput in 1995 was 3,000,000 m^3 of fuel. This is an average of some 9 million litres a day. In 1996 the average a day increased to 9,5 million litres. In 1997 the estimation is 10 million litres a day. AFS has so much storage-capacity that the throughput for a period of ten days can be guaranteed at the airport.

As it is important to know where the most energy is used, the next paragraph will discuss the bills etc.

2.2 AFS energy bills.

In table 2.1 the energy usage, the maximum amount of absorbed power (P_{max}) in a year and the bill for the different places at AFS are given.

From table 2.1 it is noticed that RijkI far out has the largest bill. The decrease at Oost is the result of the decrease in jetties which deliver fuel. Each year more fuel is delivered by DPO.

Utilities can save energy in fluid distribution operations in only a limited number of ways. Turning off lights or fans in pump stations when the stations are not occupied or using pickup trucks with small engines can marginally affect energy use. But it is only by improving the operation of pumps, which are the biggest energy users in a distribution system, that utilities can realize significant energy cost savings [Walski 1993]. Now the energy usage for the different parts will be calculated and compared.

Table 2.1: **The energy usage and bill of AFS in 1996 compared with 1995. (for more information: see Appendix B1).**

Facility	Energy Usage 1996 [1000 kWh]	Pmax 1996 [kW]	Bill 1996 [*f* 1000 Guilders]	Bill 1995 [*f* 1000 Guilders]
Oost	663	282	146.244,41	187.766,10
RijkI	1.591	1042	376.290,65	356.985,94
RijkII	827	294	163.481,33	61.478,83
Centrum	151	71	32.114,60	37.628,65
Hydrants and Pipelines			8.742,81	9.377,22

By using flowrate in relation to time printouts the total flow in a day/week or year could be estimated. From one week in February 1997 the total flow used at Schiphol, is equal to 67833 m^3 (one average day is equal to 9690 m^3. So it can be concluded that this estimation is very realistic.

If one wants to calculate the total energy usage, one should use the following formula:

$$\sum_{i=1}^{n} P_{motor\,i} \cdot T_i \qquad (2.2.1)$$

where $P_{motor\,i}$ = Power used by motor (see also paragraph 3.4);
T_i = Fraction of time at operating point i;
n = Number of operation points.

The result of this calculation is an energy usage of 29364 kWh a week (approximately 130000 kWh a month). Normally the energy usage a month at RijkI is 150000 kWh, so the pumps use about 86,6 percent of the total energy. This definitely makes them the biggest energy users at RijkI.

For the situation at Oost it is very difficult to calculate the energy usage of the pumps, because there are no flow-time printouts available. For a rough estimation the following conditions are satisfied:

- There is no pumping via the 12" (backup) pipeline to valve chamber P9a;
- At this moment two or three jetties a day will deliver the fuel at Oost. This means an average of 2,5 jetties a day and a flowrate of 3000 m^3 a day (1200 m^3 a jettie);
- The standard duty cycle of a centrifugal pump given in figure 2.1, will be used.

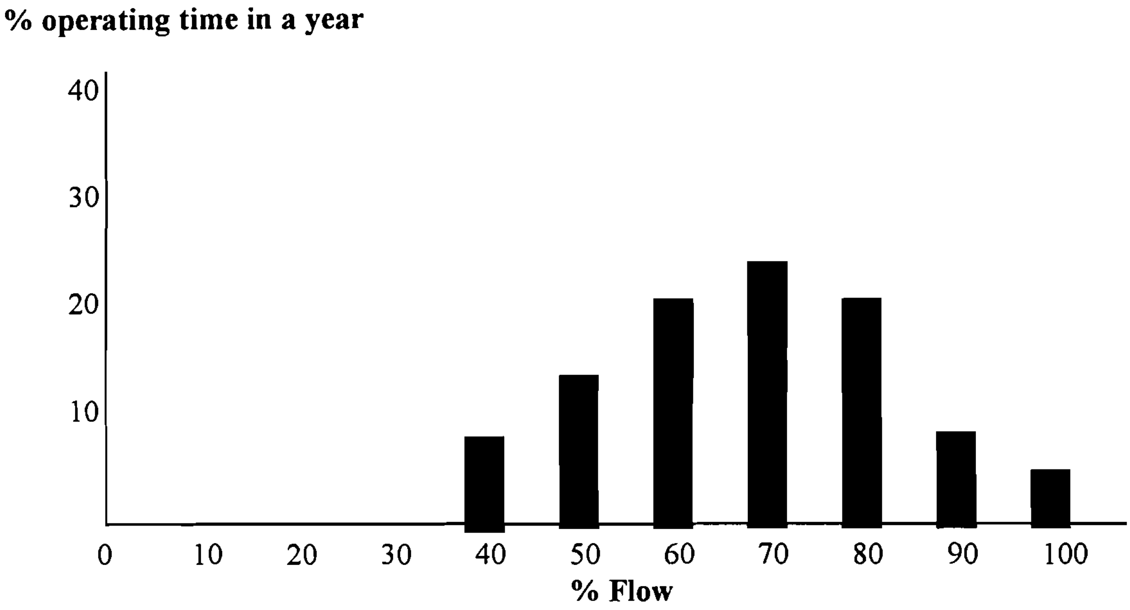


Figure 2.1: Typical centrifugal pump duty cycle:

The total estimated amount of time to empty one jettie is 2,5 hours. So 2,5 jetties will last 2,5·2,5=6,25 hours (it doesn't matter if the ships are unloaded at the same time). By using figure 2.1, the formulas given in paragraph 3.5.2 and a motor efficiency of 0,95, the total energy usage a day is 1254 kWh (37612 kWh a month). So the pumps use 75,2 percent of the total energy usage at Oost.

This percentage is smaller than that of RijkI, but as already said before, the pumping through the12" pipe has not been taken into account and it was a rough estimation.

At RijkII there are three booster pumps. An estimation for the percentage of the total energy usage for the pumping is 90.

A summary of these figures is given in table 2.2.

Table 2.2: Estimation of the percentage of the energy usage at the different places.

Facility	Pumping	Lighting	Refuelling	Air-conditioning
RijkI	86,6	8,4	n.a.	5
RijkII	90	10	n.a.	n.a.
Oost	75,2	9,8	9	6

The air-conditioning in the main building (built in 1995) is very new, so an energy saving will be very difficult to establish.

It can be concluded that the pumps are the most important energy users. So only the pumping systems will be considered.

Most utility managers would deny that their pumping stations are operated inefficiently, and they are generally justified in that assertion. It is unlikely, however, that any given pump station, although probably not extremely inefficient, is being operated at its lowest possible energy cost. The issue is really one of trying to get as close as possible to this theoretical least-cost operation without taking any unjustified risks or spending excessive money on the required hardware, software, and studies. First of all the objectives of AFS must be considered.

2.3 Utility's objectives conflict

The minimization of energy consumption is not the sole, or even the most important, goal in operating distributing systems. Rather, it is one of many objectives, some of which are directly contrary to energy minimization. Any proposed operating policy for a distribution system must be evaluated with regard to its impact on these competing objectives. Now some examples of these objectives will be given:

- ***Everyday demands must be satisfied.*** Meeting consumer (Schiphol) demands is the **number one goal** of the AFS pumping system, and saving energy should never take precedence over satisfying demands. Schiphol is not willing to go without kerosine just to save a fraction of energy cost.
- ***Emergency demands must be met.*** The primary purpose of storage fuel at Rijk is to provide kerosine at Schiphol to also meet the very high demands at days when no fuel is delivered by DPO or the tankers.
- ***The quality of the kerosine must be maintained.*** If the level in a tank remains constant, kerosine cannot be exchanged between the tank and the distribution system.
- ***Capital cost must be minimized.*** The cost of energy to overcome friction head loss in pipes can be reduced by installing large pipes so that the head loss (and thus the energy needed to overcome the head loss) is minimized [Walski, 1983]. Usually, if a pipe is sized for a velocity of less than 1,5 m/s at peak flow, the head loss for average flow will be reasonable. The marginal cost of upsizing such a pipe will generally exceed the marginal energy reduction. But it may be worthwhile in some cases, such as long pipelines in which peak and average flow are not too different, to consider pipes for energy reduction. Also one should keep in mind future expansions (for AFS this is imporatant).

It is thus clear that energy minimization is only a secondary goal in the AFS distribution system operation, and methods of reducing energy usage are limited. Nevertheless, there are ways to reduce energy costs and still meet many objectives of the system operation.

2.4 Causes of inefficiency are numerous

Because there is no single reason why pumping systems are operated less than optimally, there is no single, simple approach to minimizing pumping energy costs. Instead, there are a myriad reasons why pumping stations operate at a higher-than-optimal cost, including:

- pumps are incorrectly selected;
- pumps have worn out or have clearances set incorrectly;
- capacity in the transmission-distribution system is limited;
- operation of pressure (hydropneumatic) tanks is inefficient;
- pumps or valves cannot be controlled automatically or remotely;
- a penalty results from time-of-day or seasonal energy pricing;
- demand or capacity power charges are not understood;
- operator error occurs;
- pump control strategies are less than optimal.

Most of these problems will not be noticed unless the utility actively looks for energy waste.

2.5 Several factors affect energy use

Energy consumption is directly proportional to the discharge, head, and inversely proportional to the efficiency of the pumps. The problem is complicated, however, by the fact that the efficiency is itself a function of the discharge.

$$\sum_{i=1}^{n} k \cdot Q_i \cdot H_i \cdot p_i \cdot T_i / e_i \qquad (2.5.1)$$

where	Q_i =	Discharge rate operating point i	[m^3/h];
	H_i =	Head at operating point i	[m];
	p_i =	Price of energy at operating point i	[guilders/kWh];
	T_i =	Fraction of time at operating point i	[h];
	e_i =	Wire-to-kerosine efficiency at operating point i	[%];
	n =	Number of operation points.	

From equation 2.5.1 it is clear that energy usage can be reduced by

1. *decreasing the volume of water pumped.* Unfortunately AFS does not have much control over the volume of kerosine pumped to Schiphol. Saving energy by not meeting demands is not acceptable;
2. *decreasing the head against which it is pumped.* Reducing the total dynamic head (TDH) against which the pump will work will reduce energy consumption. The easiest way to accomplish this is to keep distribution storage tanks (not available at AFS) less than full and suction storage tank as full as possible [Rehis et al., 1984]. This will save energy provided that the efficiency of the operating point with these new tank levels does not deviate much from the best efficiency point for the pump;

This will save energy provided that the efficiency of the operating point with these new tank levels does not deviate much from the best efficiency point for the pump;
3. *increasing the efficiency of the pumps.* Utilities need to check periodically to ensure that pumps are operating near their best efficiency point. A utility can mistakenly conclude that because a pump is supplying adequate pressure and flow to a given area, it is working well. In fact, the pump may be robbing the utility of thousands of dollars each year in the form of unnecessarily high energy bills.

2.6 Recommendations about the energy bills

The different properties of AFS all have different energy bills. As already said before AFS is not only charged for the actual energy consumed but for the energy demand-capacity required. Capacity charges relate to the electric power utility's assessment of maximum potential load, whereas demand charges are based on the actual peak load experienced by the utility during a predetermined monitoring period.

The contract power is the maximum power peak that AFS may use without extra charges. The power guarantee is 60% of this maximum power peak and this is the minimum that AFS should pay, also if the actual power peak absorbed is less. These figures are given in table 2.3 for the most important properties of AFS.

Table 2.3: Contract power between AFS and the energy company for different properties of AFS.

AFS Property	RijkI	RijkII	Oost
contract power [kW]	960	490	400
power guarantee [kW]	576	294	240

There are four different rates (operation times) for the energy and power. These are calculated as follows:

Rate = (Total amount of kWh used) / (maximum peak in a year) [h] (2.6.1)

This results in:
- RijkI: 1527 hours (rate II, see table 2.4).
- RijkII: 3957 hours (rate III, see table 2.4)
- Oost: 2350 hours (rate III, see table 2.4)

Now the review of the energy bills of 1996 for the different properties given in the both tables mentioned above will be dealt with.

Table 2.4: Different rates for calculating the energy prices for four different operation times.

Rates	I operation time: ≤ 1000 hours	II operation time: 1000 - 2000 h	III operation time: 2000 - 4000 h	IV operation time: ≥ 4000 h
kW costs a month guilders/kW				
nov - feb:	6,20	9,60	13,50	17,10
ma - oct:	12,40	19,20	27,-	34,20
kWh costs cents/kWh	11,4	6,8	4,0	2,5
Fuel costs cents/kWh				
jan - ma:	5,61	5,61	5,61	5,61
apr - sept:	5,35	5,35	5,35	5,35
oct - dec:	5,46	5,46	5,46	5,46
fixed costs guilders a month	90	90	90	90

2.6.1 RijkI

The maximum power peak a month varied in 1996 for RijkI between 744 kW (january) and 1042 kW (august). In august more power (1042 kW) was taken than the contract power (960 kW). This may happen **once** a year. When it happens two times or more the contract power is raised to a higher amount. For example:
There a two times in a year when the contract power is exceeded (1042 kW and 1000 kW). The contract power will now be raised to 1050 kW (rounded to amounts of tens). It will cost for every extra kVA *f* 50,-. An increase of 90 kW as mentioned in the example above is equal to (amount of kW)/(power factor) = 90/0.85 = 106 kVA, what results in a payment of ***f* 5300,-.**

The example mentioned above almost happened and it will probably happen in 1997, because more fuel will be pumped.

2.6.2 RijkII

At RijkII the maximum peak measured in 1996 did not exceed 209 kW. From table 2.3 it is obvious that AFS paid too much for every month (power guarantee = 294 kW). AFS paid in 1996 about *f* 17000,- (excl. tax), which means ***f* 20000,-** (incl. tax), too much.

In february 1997 the contract power was changed to 320 kW (power guarantee of 192 kW) so AFS now don't longer pays too much.

2.6.3 Oost

The power peak at Oost varied between 252 (july) - 282 (february) kW in 1996. So the contract power at this place is good.

It is noticed that the power peak has decreased in the last years, because of the decrease in jetties every year which arive at Oost, and the resulting decrease in pumping energy usage.

By reviewing the energy bills very good, one can see that also at Oost too much has been paid in 1996:
From table 2.4 (2350 hours) it is obvious that Oost should have paid the costs for rate III. The energy bill, however, is calculated by rate IV. This is caused by the fact that a property is divided in a rate from the operation time in the last year. So at Oost the operation time in 1995 should have been more than 4000 hours.

This costed AFS in 1996 extra: ***f* 17000,-**

It is obvious that it is very important to study the energy bills very good. One can save a lot of money.

By calculating the energy savings these values will also be used. One should also keep these in mind when giving recommendations.

Chapter 3

Centrifugal Pumps

3.1 Introduction

The twelve hydrant pumps which transport the Jet A-1 to Schiphol centre, are all centrifugal pumps. In these pumps the energy transfer is accomplished by changing the state of motion of the liquid. The magnitude of this change depends upon the dimensions and shape of the waterways and on the operating speed of the impeller. For a given pump geometry and operating speed, the velocities of the liquid vary with the flow rate. Consequently, centrifugal pumps develop different heads at different flow rates. Similary, efficiency, power requirements, and suction capability also vary with flow rate.

At Rijk and Oost the Sulzer CZ 100-315 centrifugal pump is used. The pumps of series CZ are horizontal single-stage single suction centrifugal process pumps. Their dimensions and performances are in conformity with DIN 24256 / ISO 2858.

The impeller diameter of the Sulzer CZ 100-315 is 315mm.

In the following paragraphs the centrifugal pump will be discussed in general and some important parameters for the Sulzer CZ 100-315 will be calculated.

3.2 Performance charts

3.2.1 In general

In common parlance, the performance-features of a centrifugal pump is described by specifying the head developed by the pump at a given flow rate. In practice, however, a centrifugal pump rarely operates against a certain given head, or delivers a constant amount of liquid. Most pumps must operate over a certain range of heads and flows. Consequently, it is often very important to know how the performance features of a pump change with the flow rate.

When a new pump is developed by a manufacturer, it is tested under controlled conditions. The results of these tests are then plotted as curves on what are commonly known as *performance charts*. Figure 3.1 shows a typical performance chart. The following curves are given in figure 3.1:

1. The curve marked QH which shows how the developed head changes with the flow rate;
2. The curve marked HP which represents the power consumed at different flows (horsepower);
3. The curve marked EFF which shows the ratio between the actual amount of power added to the liquid and the power consumed by the pump at the given flow rate (efficiency).
4. The curve marked NPSH (Net Positive Suction Head) which shows the minimum head required at the suction nozzle of the pump to avoid cavitation.

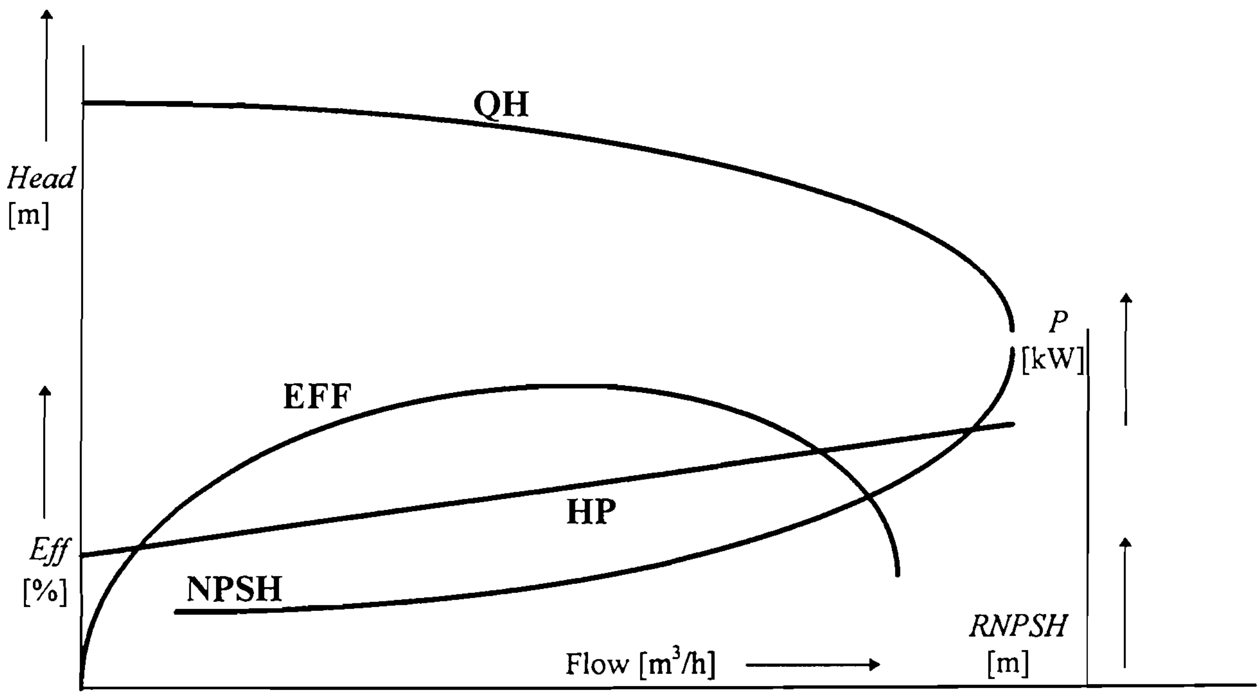


Figure 3.1: A typical performance chart of a centrifugal pump. P = power; RNPSH = required NPSH; Eff = Efficiency

Under normal operating conditions, a pump is expected to demonstrate the same performance characteristics in the field as shown in the performance chart. If it doesn't, something is probaly wrong with either the pump or the pumping system.

In a pump with a flat QH curve, the power consumption usually increases continuously with the flow rate. This may lead to overloading of the power supply in the event of a rupture in a discharge pipe or some similar accident. A pump with a very steep QH curve, however, may overload the driver at reduced flow rates.

One of main advantages of higher efficiency is the reduction in operating costs. It has been estimated that an average increase of 3% in efficiency can repay the total cost of the pump within a year [Yedidiah, 1996]. But cost savings are not the only advantage of higher efficiencies: a more efficient pump may need a smaller motor, thus requiring less floor space. Higher efficiency also means less power is spent on friction and other activities that may cause excessive wear of the parts.

The following equations give the relation between the head, flow rate, power and speed:

- The flow rate in relation to the speed:

$$\frac{q_{v_1}}{q_{v_2}} = \frac{n_1}{n_2} \tag{3.2.1.1}$$

- The head in relation to the speed:

$$\frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{3.2.1.2}$$

- The power absorbed in relation to the speed:

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{3.2.1.3}$$

3.2.2 Sulzer CZ 100-315

The centrifugal pumps used at Rijk and Oost are Sulzer CZ 100-315 pumps. The performance charts of this pump are given in figure 3.2.

The characteristic curves indicate the behaviour of the pump under changing operating conditions. The head H, power input P and efficiency η at constant speed n are plotted against the flow rate Q.

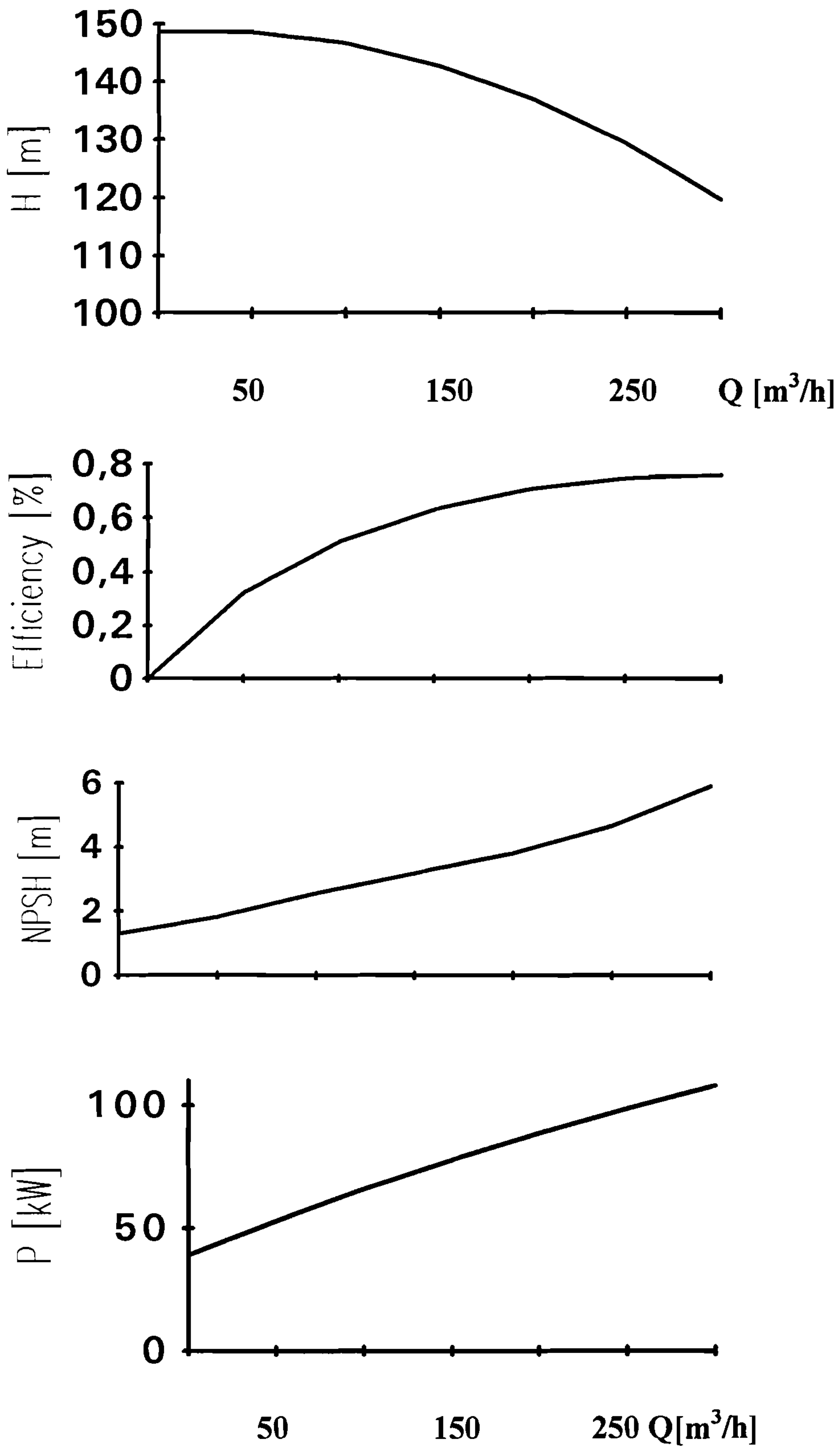


Figure 3.2: **Performance charts of the Sulzer CZ 100-315 pump. Speed n = 2950 rpm. Density $\rho = 840$ kg/m^3.**

The design point (maximum efficiency) has the following values:

- $\eta = 74{,}9\ \%$
- $H = 127{,}0$ m
- $Q = 260{,}0\ m^3/h$
- $P = 100{,}9$ kW
- $n = 2950$ rpm
- NPSH = 4,9 m

3.3 Specific Speed

3.3.1 In general

The design and performance of a pump is greatly dependent on the relation between its operating speed, the flow rate for which it has been designed, and the developed head. In practice, this relation is defined as the *specific speed*. In the United States, its magnitude is expressed by the equation

$$N_S = n \cdot \frac{Q^{0.5}}{H^{\frac{3}{4}}} \tag{3.3.1.1}$$

where	n	=	Operating speed [rpm]
	Q	=	Flow rate [Gpm]
	H	=	Head [feet]

When expressing the specific speed in metric units, the flow rate Q is usually expressed in $[m^3/sec]$ and the head H in meters. In that case, the relationship between the specific speeds expressed in U.S. units and metric units is

$$N_s(\text{U.S.}) = 51.64 \cdot N_s(\text{metric}). \tag{3.3.1.2}$$

For the metric units equation 3.3.1.1 can also be used. Only then with:

n	=	Operating speed [rpm]
Q	=	Flow rate $[m^3/s]$
H	=	Head [m]

The N_S(metric) varies between 10 and 500 $m^{3/4}/s^{3/2}$. A small value of N_S has a high head and a small flow.

3.3.2 Sulzer CZ 100-315

Now the magnitude of the specific speed at the design point can be calculated (equation 3.3.1.1):

$$N_S \text{ (metric)} = (2950/127^{0.75}) * \sqrt{(260/3600)} = 21 \text{ m}^{3/4}/\text{s}^{3/2}$$

It is noticed that this type of pump has a low specific speed.

3.4 Power, losses and efficiencies

The effective power transmitted to the liquid (P_Q) is given in equation (3.4.1) [Wijers, 1991]:

$$P_Q = \frac{Q \cdot H \cdot \rho \cdot g}{36 \times 10^5} \quad \text{[kW]} \tag{3.4.1}$$

where Q = The useful flow rate (at the pump delivery nozzle) [m³/h]
H = Head [m]
ρ = Density of the liquid [kg/m³]
g = Gravitational acceleration [m/sec²]

The impeller flow $\mathbf{Q_I}$ generally comprises three components:
1. the useful flow rate: $\mathbf{Q}$;
2. the leakage flow rate (through the impeller sealing rings): $\mathbf{Q_L}$;
3. the balancing flow rate (for balancing axial thrust): $\mathbf{Q_B}$;

The volumetric efficiency η_v is

$$\eta_v = \frac{Q}{Q + Q_L + Q_E} \tag{3.4.2}$$

The power input required at the pump drive shaft is defined as [Sulzer, 1994]:

$$P = \frac{Q \cdot H \cdot \rho \cdot g}{\eta_v \cdot \eta_h} + P_d + P_m + P_b \tag{3.4.3}$$

where P_d = Disc friction losses (impeller side discs, labyrinths);
P_m = Mechanical losses (bearings, seals);
P_b = Frictional losses in the balancing device (disc or piston);
η_h = Hydraulic efficiency.

Pump efficiency (η) is defined as the ratio of the pumping power (useful hydraulic power) (P_Q) to the power input (P):

$$\eta = \frac{P_Q}{P} \tag{3.4.4}$$

or expressed in the form of individual efficiencies:

$$\eta = \eta_{hR} \cdot \eta_v \cdot \eta_m \tag{3.4.5}$$

where η_{hR} is the combination of the hydraulic losses and friction losses:

$$\eta_{hR} = \eta_h \cdot \left(1 - \frac{P_d + P_m}{P \cdot \eta_m}\right) \tag{3.4.6}$$

and η_m is the mechanical efficiency and is defined by:

$$\eta_m = 1 - \frac{P_m}{P} \tag{3.4.7}$$

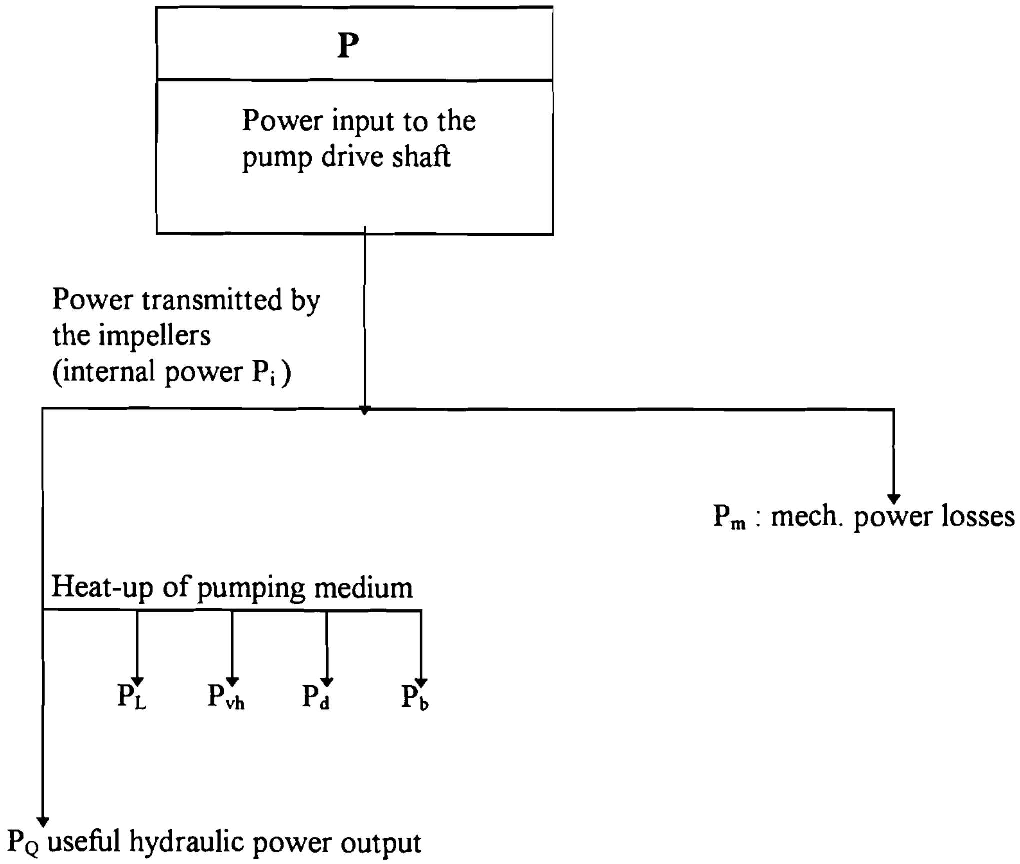


Figure 3.3: The different losses in a pump.
where: P_{vh} = the hydraulic power losses = $\rho \cdot Q \cdot g \cdot H \cdot (1/\eta_h - 1)$
$P_L = (Q_L + Q_B) \cdot \rho \cdot g \cdot H / \eta_h$

The internal efficiency incorporates all losses leading to heating of the pumping medium, such as:

- hydraulic losses
- disc friction losses
- leakage losses including the balancing flow, if the latter is returned to the pump intake as is normally the case

The total situation is given in figure 3.3.

3.5 Estimation of pump performance characteristics

3.5.1 General

The impeller (one or several) is the working part of the pump. It is mounted on a shaft and driven by the motor coupled to the pump. On radial-flow pumps, the impeller is built up out of surfaces of revolution and is divided into channels by fixed blades whose surfaces are either cylindrical or have a double curvature. The fluid is led through these channels and, according to Bernoulli's equation, increases its pressure and motion (kinetic) energy; once past the channels, the fluid is led through the stator, where most of the motion (kinetic) energy is converted into position (potential) energy.

The motion of an elementary fluid particle within the strict boundaries set by the rotor of a radial-flow pump (see figure 3.4) will now be considered [Ionel, 1986].

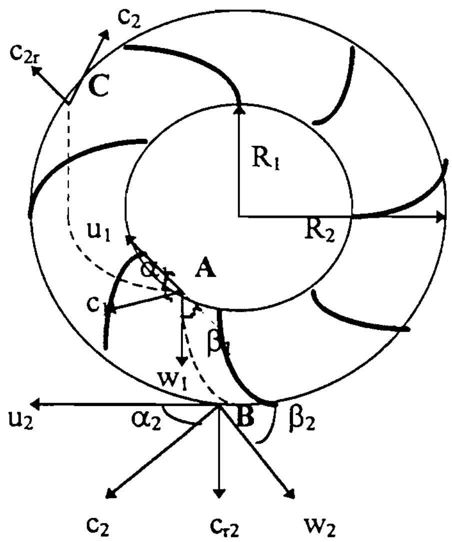


Figure 3.4: Notation for deriving the pump fundamental operational equations.

α_1, α_2: Angles between c and $u = r\omega = \pi \cdot D \cdot n/60$

β_1, β_2: Relative angles between w and u

R_1, R_2: Current radius

It is assumed that the liquid moves freely inside the pump (without friction), following paths that are parallel to the pump impeller blades. This condition can only be achieved if the blades have no thickness and are infinite in number. It is also assumed that the liquid motion is laminar and continuous.

The fluid particle entering the impeller at point *A* moves, under the action of centrifugal forces, in a translation motion from the centre towards the periphery, at a relative velocity; at the same time, it acquires a rotational motion, along with the impeller, at an angular velocity ω, and a linear velocity (the peripheral velocity) u, both velocities growing as the particle moves away from the impeller centre. The particle arriving at the impeller periphery (point *C*) is thrown into the stator at the absolute velocity c_2. The trajectory in space of the absolute motion of the particle is determined by curve *AC*, and that of the motion relative to the impeller blades by curve *AB*.

The theorem of the momentum between the sections corresponding to points *A* and *B* will now be applied. According to this theorem, the momentum torque of the external force is equal to the variation in the moment:

$$T = m \cdot c_2 \cdot r_2 \cdot \cos\alpha_2 - m \cdot c_1 \cdot r_1 \cdot \cos\alpha_1 \qquad (3.5.1.1)$$

where $T = A / \omega$ is the moment of the external force;
A is the mechanical work produced by the external force.

It is obvious that mechanical work *A* leads to an increased energy of particle motion, that is further converted into pressure energy. Thus, neglecting the losses in the machine, the theoretical pumping head H_{theor} is related to the mechanical work by the expression

$$A = m \cdot g \cdot H_{theor} \qquad (3.5.1.2a)$$

$$T = \frac{m \cdot g \cdot H_{theor}}{\omega} \qquad (3.5.1.2b)$$

By combining (3.5.1.1) and (3.5.1.2) and bearing in mind that $u = r \cdot \omega$, the equation for the theoretical pumping head in the case of an impeller with an infinite number of blades is obtained

$$H_{theor} = \frac{u_2 \cdot c_2 \cdot \cos\alpha_2 - u_1 \cdot c_1 \cdot \cos\alpha_1}{g} \qquad (3.5.1.3)$$

And, because in practice the liquid enters the impeller at an angle $\alpha_1 = 90°$, equation 3.5.1.4 is obtained

$$H_{theor} = \frac{u_2 \cdot c_2 \cdot \cos\alpha_2}{g} \qquad (3.5.1.4)$$

Actually, the real pumping head of a pump is lower than the theoretical head because pressure losses occur and, at the same time, the velocities are non-uniformly scattered across the two sided-surfaces of the impeller (caused by the recycling of one part of the fluid). As a consequence, a pressure coefficient k_h is introduced, whose expression is

$$k_h = \eta_h \cdot k_c \cdot \frac{c_2 \cdot \cos\alpha_2}{u_2} \tag{3.5.1.5}$$

where η_h = The hydraulic efficiency;
k_c = The recycling coefficient.

The design parameters of a pump (i.e. the impeller geometry, the number of blades etc.) are determining factors for the pressure coefficient. If this coefficient, which is always lower than unity, is included in equation 3.5.1.4, the expression for the pumping head will be

$$H = k_h \cdot \frac{u_2^2}{g} - k_h \cdot \frac{r_2^2}{g} \cdot w^2 \tag{3.5.1.6}$$

From equation 3.5.1.6 it can be concluded that the pumping head developed by the pump is proportional to the square of the angular rotational speed.

The theoretical capacity of the pump Q_{theor} is obtained by multiplying the impeller output cross-sectional area by the radial velocity component of the fluid streaming out of the impeller (see fig. 3.4). Thus

$$Q_{theor} = \pi \cdot d_2 \cdot b \cdot c_{2r} \tag{3.5.1.7}$$

where b denotes the width of the pump impeller.

By means of the velocity triangle at the outlet (see fig. 3.4), equation 3.5.1.8 is obtained

$$c_2 \cdot \cos(\alpha_2) = u_2 - c_{2r} \cdot \cot(\beta_2) \tag{3.5.1.8}$$

Substituting equation 3.5.1.8 in equation 3.5.1.4 and introducing c_{2r} from equation 3.5.1.7, an idealized relation is obtained relating the pumping head and the capacity, i.e. the theoretical characteristic H(Q) of the pump

$$H_{theor} = \frac{u_2^2}{g} - \frac{u_2}{g} \cdot \frac{\cot\beta_2}{\pi \cdot d_2 \cdot b} \cdot Q_{theor} \tag{3.5.1.9}$$

The linear velocity u_2 is expressed as a function of the rotation speed n, in rpm:

$$u_2 = \frac{\pi \cdot d_2 \cdot n}{60}$$

and substituting

$$A_1 = \frac{\pi^2 \cdot d_2^2}{3600 \cdot g}$$

$$B_1 = \frac{\cot \beta_2}{60 \cdot g \cdot b}$$

the result will be

$$H_{theor} = A_1 \cdot n^2 - B_1 \cdot n \cdot Q_{theor} \qquad (3.5.1.10)$$

The last equation expresses a linear relationship between the pumping head and the capacity. It has an idealized character because, like equation 3.5.1.4, it excludes the losses due to recycling processes, which are actually independent of the discharge pipe flow rate. This losses can be expressed through a component C_1n^2. This results in

$$H'_{theor} = (A_1 - C_1)n^2 - B_1 \cdot n \cdot Q \qquad (3.5.1.11)$$

Likewise, it must be remembered that relation 3.5.1.4 was derived for the optimal performance conditions of the pump, i.e. for a rated flow Q_r. When $Q < Q_r$, the fluid stream (flow) comes off the blade black-wall, while when $Q > Q_r$, the fluid stream comes off the blade front-wall. Pressure losses due to the fluid stream pull-off have a vital influence on the pump speed-torque characteristic.

If in (3.5.1.10), which is the expression for the theoretical characteristic H(Q) of the pump, due consideration is given to losses outlined in figure 3.5 that take place inside the pump, then the real characteristic H(Q) is obtained as a second order expression

$$H = A_2 \cdot n^2 + B_2 \cdot n \cdot Q + C_2 \cdot Q^2 \qquad (3.5.1.12)$$

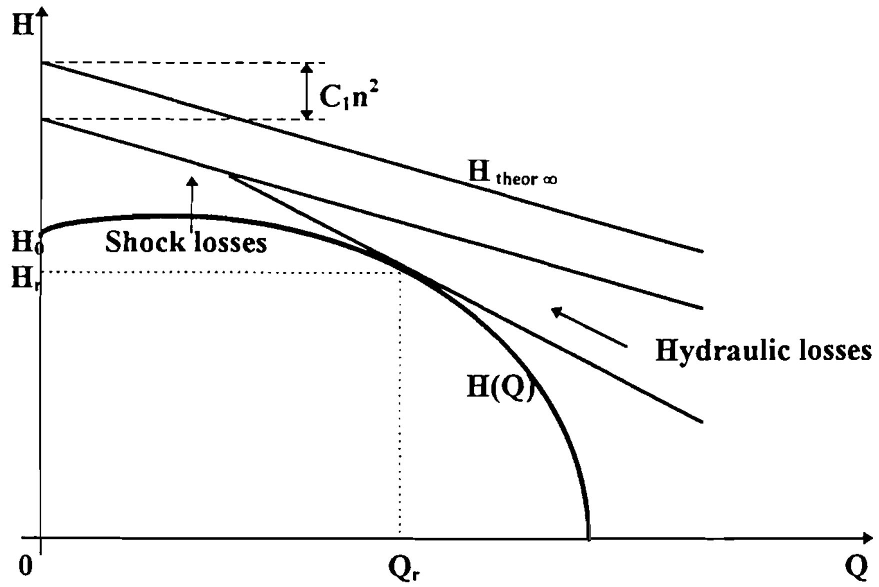


Figure 3.5: Pump head-capacity characteristic.

By multiplying both terms of equation 3.5.1.10 by ρ·g·Q, one gets (3.5.1.13) that expresses the dependence between power and capacity, i.e. the P(Q) characteristic of the turbopump, thus

$$P' = A_3 \cdot n^2 \cdot Q - B_3 \cdot n \cdot Q^2 \qquad (3.5.1.13)$$

This relation doesn't take into consideration either the recycling or the mechanical losses, for neither of them is a function of the external capacity of the pump. Indeed, the recycling losses are produced as a consequence of the parasite recycling, between the rotor and the stator, of one portion of the flow, while the mechanical losses are the consequence of friction between bearings and seals, and between the rotor disc and the cycled fluid.

If one takes these losses in consideration, along with the fact that they are proportional to the third power of the speed, the equation of the real characteristic of the pump power requirements as a function of the capacity results in the form

$$P = A_3 \cdot n^2 \cdot Q - B_3 \cdot n \cdot Q^2 + D_3 \cdot n^3 \qquad (3.5.1.14)$$

The pump efficiency η can be expressed from (3.4.1) and (3.4.4) as

$$\eta = \frac{\rho \cdot Q \cdot H}{367000 \cdot P} \qquad (3.5.1.15)$$

where Q = The capacity [m^3/h];
H = The pumping head [m];
P = The mechanical power used [kW];
ρ = The density of the liquid [kg/m^3].

Substituting (3.5.1.14) in (3.5.1.15) results in

$$\eta = \frac{\rho}{367000} \cdot \frac{Q \cdot H}{A_3 \cdot n^2 \cdot Q - B_3 \cdot n \cdot Q^2 + D_3 \cdot n^3} \qquad (3.5.1.16)$$

Taking account of the pumping head H as expressed by relation 3.5.1.12, one can write for the real characteristic of the efficiency, as a function of the capacity, the relation

$$\eta = \frac{\rho}{367000} \cdot \frac{A_2 \cdot n^2 \cdot Q + B_2 \cdot n \cdot Q^2 + C_2 \cdot Q^3}{A_3 \cdot n^2 \cdot Q - B_3 \cdot n \cdot Q^2 + D_3 \cdot n^3} \qquad (3.5.1.17)$$

In figure 3.6 the range of rational use for a radial-flow pump is given.

In the next section the characteristics H(Q) and P(Q) of the Sulzer CZ 100-315 will be determined.

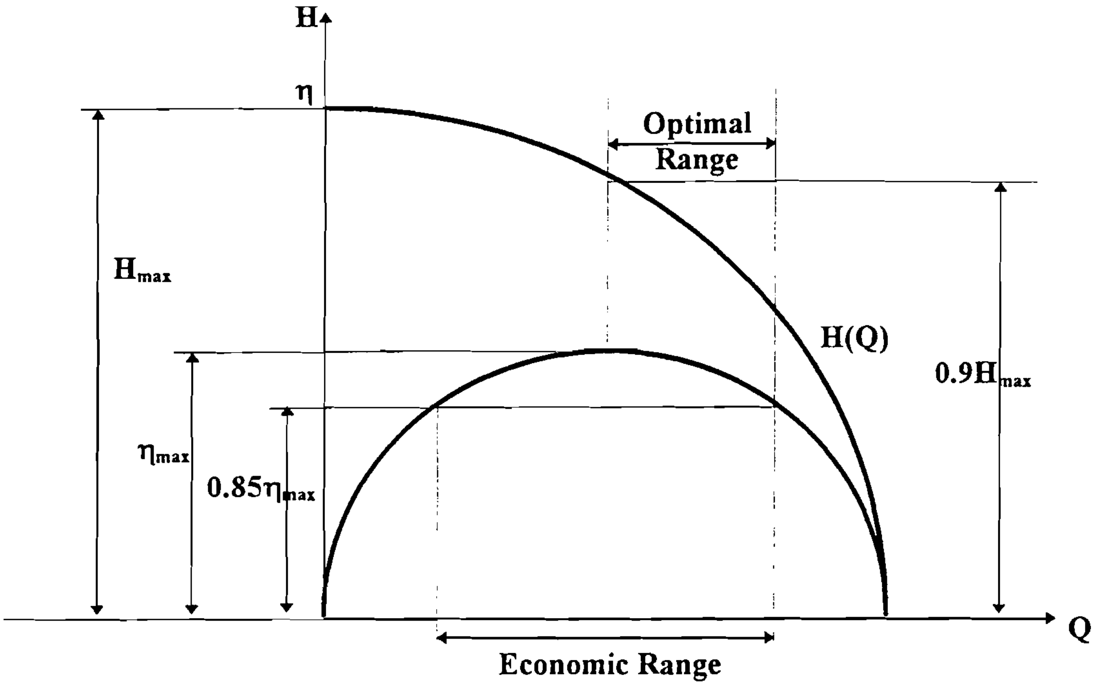


Figure 3.6: Range of rational use for a radial-flow pump.

3.5.2 Determination of the characteristics of the Sulzer CZ 100-315

First the characteristics H(Q) and P(Q) will be determined. Then the η(Q) can be easily calculated.

(1) *Determination of characteristic H(Q)*. This is done with relation (3.5.1.12); coefficients A_2, B_2 and C_2 included within this relation are determined for three points, namely

$Q = 0\ m^3/h$; $H = 148.4\ m$; $P = 39.0\ kW$
$Q = 100\ m^3/h$; $H = 146.1\ m$; $P = 65.6\ kW$
$Q = 260\ m^3/h$; $H = 127.0\ m$; $P = 100.9\ kW$

Coefficient A_2 is found thus

$$A_2 \cdot n^2 = 148.4; \qquad A_2 = 148.4 / 2950^2 = 1.71 \cdot 10^{-5}$$

Substituting in relation 3.5.1.12 the value obtained for A_2, as well as values Q and H for the other two points, results in

$$C_2 = -3.71 \cdot 10^{-4}; \qquad B_2 = 4.81 \cdot 10^{-6}$$

Thus:

$$H = (171 \cdot n^2 + 48.1 \cdot n \cdot Q - 3710 \cdot Q^2) \cdot 10^{-7} \tag{3.5.2.1}$$

(2) *Determination of characteristic P(Q).* This is done with relation (3.5.1.14). Coefficients A_3, B_3 and D_3 in the relation are also determined for three points, namely

$$D_3 \cdot n^3 = 39; \quad D_3 = 39/2950^2 = 1.52 \cdot 10^{-9}$$

$$A_3 = 3.26 \cdot 10^{-8}; \quad B_3 = 5.933 \cdot 10^{-8}$$

Thus:

$$P = (326 \cdot n^2 \cdot Q - 593.3 \cdot n \cdot Q^2 + 15.2 \cdot n^3) \cdot 10^{-10} \tag{3.5.2.2}$$

(3) *Determination of characteristic η(Q).* This is done with relation 3.5.1.15 for $\rho = 840\ kg/m^3$.

Thus:

$$\eta = 2{,}29 \cdot 10^{-3} \cdot \frac{(171 \cdot n^2 \cdot Q + 48{,}1 \cdot n \cdot Q^2 - 3710 \cdot Q^3) \cdot 10^{-7}}{(326 \cdot n^2 \cdot Q - 593{,}3 \cdot n \cdot Q^2 + 15{,}2 \cdot n^3) \cdot 10^{-10}} \tag{3.5.2.3}$$

For **n = 2950 rpm** the resulting values are:

$$H = 148.8 + 0.0142 \cdot Q - 3.71 \cdot 10^{-4} \cdot Q^2 \tag{3.5.2.4}$$

$$P = 0.2837 \cdot Q - 1.75 \cdot 10^{-4} \cdot Q^2 + 39 \tag{3.5.2.5}$$

$$\eta = 2{,}29 \cdot 10^{-3} \cdot Q \cdot \frac{148{,}8 + 0{,}0142 \cdot Q - 3{,}71 \cdot 10^{-4} \cdot Q^2}{0{,}2837 \cdot Q - 1{,}75 \cdot 10^{-4} \cdot Q^2 + 39} \tag{3.5.2.6}$$

3.6 Parallel-connection of pumps

A configuration where the pumps are parallel-connected has the main purpose of increasing the pumping plant capacity. Now two parallel-connections will be discussed:
1. parallel-connection of identical pumps;
2. parallel-connection of non-identical pumps.

3.6.1 Parallel-connection of identical pumps.

The characteristic of this configuration is found by successively summing up the outputs of the two pumps for various pumping heads. In the case under consideration, the abscissae of curves H_1 and H_2 are doubled to get the characteristic of the combined structure (see figure 3.7).

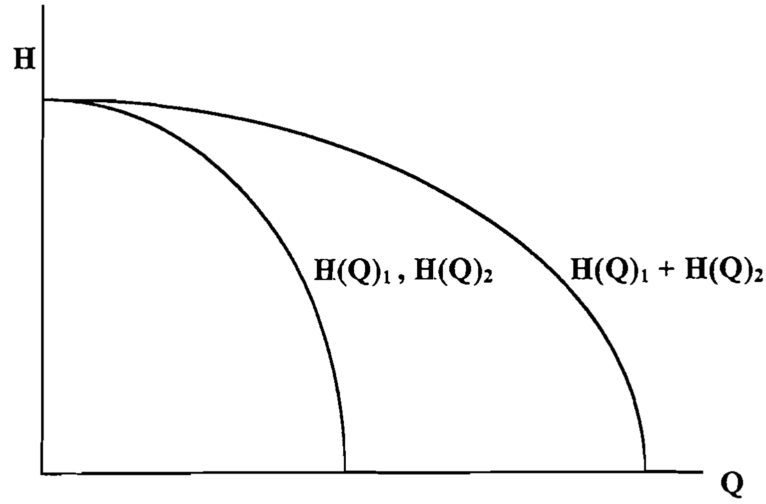


Figure 3.7: Parallel-connection of two identical pumps ($n_1=n_2$).

The efficiency of the parallel-connection is called the ratio of the useful power of the configuration to the power requirements of the two pumps.

$$\eta_{cpar} = \frac{\rho \cdot Q_{tot} \cdot H}{P_1 + P_2} \tag{3.6.1.1}$$

Because of their connection, the two pumps will have the duty point at the level of pumping head H, the output discharged by each pump will be $Q_{1,2} = 1/2 \cdot Qtot$, and if the total efficiency is the same for both pumps, then

$$P_1 = P_2 = \frac{\rho \cdot Q_{1,2} \cdot H}{\eta} = \frac{1}{2} \cdot \frac{\rho \cdot Q_{tot} \cdot H}{\eta} \tag{3.6.1.2}$$

that is

$$\eta_{cpar} = \frac{Q_{tot} \cdot H}{\frac{1}{2} \cdot \frac{\rho \cdot Qtot \cdot H}{\eta} + \frac{1}{2} \cdot \frac{\rho \cdot Q_{tot} \cdot H}{\eta}} = \eta \tag{3.6.1.3}$$

Therefore, whenever two identical pumps are parallel-connected, the coupling efficiency will be the same as the total efficiency of the pump.

3.6.2 Parallel-connection of non-identical pumps

The characteristic of this configuration is also found by successively summing up the outputs of the two pumps for various pumping heads. This is illustrated in figure 3.8.

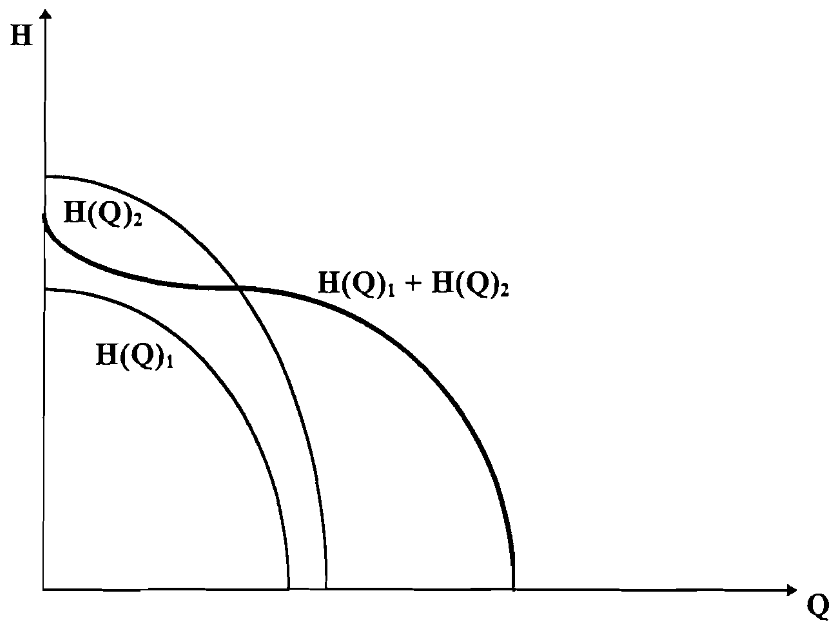


Figure 3.8: Parallel-connection of two non-identical pumps.

It is noticed that for certain performance ranges situated above the maximum pumping head of the smaller pump (pump 1), the characteristic of the whole configuration lies below the characteristic of the bigger pump (pump 2). This is a consequence of the fact that one of the two pumps operates in braking mode (that is, flybacks will develop in the smaller pump). This situation should be avoided.

The efficiency of the configuration is found by determining each duty point of each separate pump and then obtaining the both efficiencies on the efficiency curves. Hence

$$\eta_{cpar} = \frac{\rho \cdot Q_{tot} \cdot H_{tot}}{\frac{\rho \cdot Q_1 \cdot H_1}{\eta_1} + \frac{\rho \cdot Q_2 \cdot H_2}{\eta_2}} = \frac{Q_{tot}}{\frac{Q_1}{\eta_1} + \frac{Q_2}{\eta_2}} \tag{3.6.2.1}$$

In conclusion, two or more non-identical parallel-connected pumps perform as one single pump whose characteristic would be $H_1(Q) + H_2(Q)$ only if $H \leq \min(H_{1max}, H_{2max})$.

3.7 Torque/speed curves

3.7.1 General

Normally, the centrifugal pump starting torque is so low that it doesn't require special consideration. Driving units like steam, gas and water turbines, have very high driving torques. The difference between driving and pump torque is sufficient to accelerate a centrifugal pump to its operating speed in a very short time.

Depending upon their type and size, electric motors have various torque characteristics (see paragraph 4.5). With high power units in particular, start-up must therefore be investigated thoroughly at the planning stage. At around 80% of the speed the torque of an electric motor may drop to the pump torque. Acceleration torque is then nil, and the group becomes locked into this speed. As a result, the motor takes up a high current, with the risk of destroying the windings.

Pump torque T is calculated as follows:

$$T = 9545 \cdot \left(\frac{P}{n}\right) \qquad [\text{Nm}] \qquad (3.7.1.1)$$

where P = Pump power input [kW];
n = Operation speed [rpm].

In accordance with the affinity law, the torque varies as the square of the speed:

$$T_2 = T_1 \cdot \left(\frac{n_2}{n_1}\right)^2 \qquad (3.7.1.2)$$

To achieve initial pump break-off, a starting torque between 10 and 25% of pump torque at best efficiency point is normally required because of the static friction of the moving parts involved.

The torque curve depends very much on specific speed. Pumps with lower specific speed have a torque characteristic that rises with flow rate, whereas pumps with high specific speed have a characteristic that falls, as delivery increases.

3.7.2 Sulzer CZ 100-315

The pump torque at the design point is (equation 3.7.1.1):

T = 9545*(100,9/2950) = 326,5 Nm

In figure 3.9 the torque-speed curve for the Sulzer CZ 100-315 is given.

3.8 Start-up

In this paragraph three different start-ups for a centrifugal pump will be given [Pumping manual, 1996]:

1. Start-up against closed valve;
2. Start-up against closed non-return valve with the delivery valve open;
3. Start-up with delivery valve open but no geodetic head.

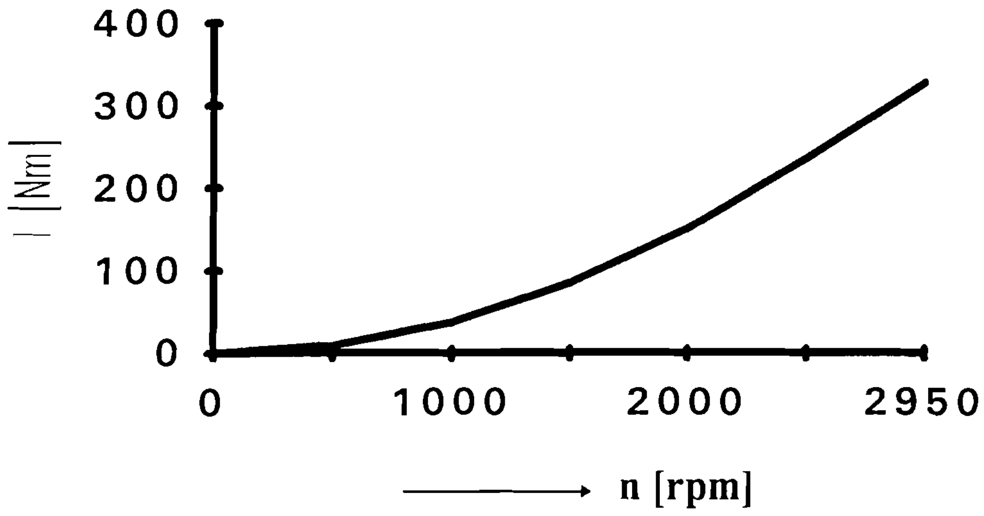


Figure 3.9: Torque-speed curve of the Sulzer CZ 100-315.

3.8.1 Start-up against closed valve

In figure 3.10 the start-up of a centrifugal pump against a closed valve is plotted. With comparatively high ratings, the pump must start with a minimum flow valve opened, due to the risk that liquid will evaporate inside the pump. As soon as operating speed is reached, the discharge valve is opened. This method is possible only on pumps with low specific speed, where the power input and torque with closed valve are less than at the duty point.

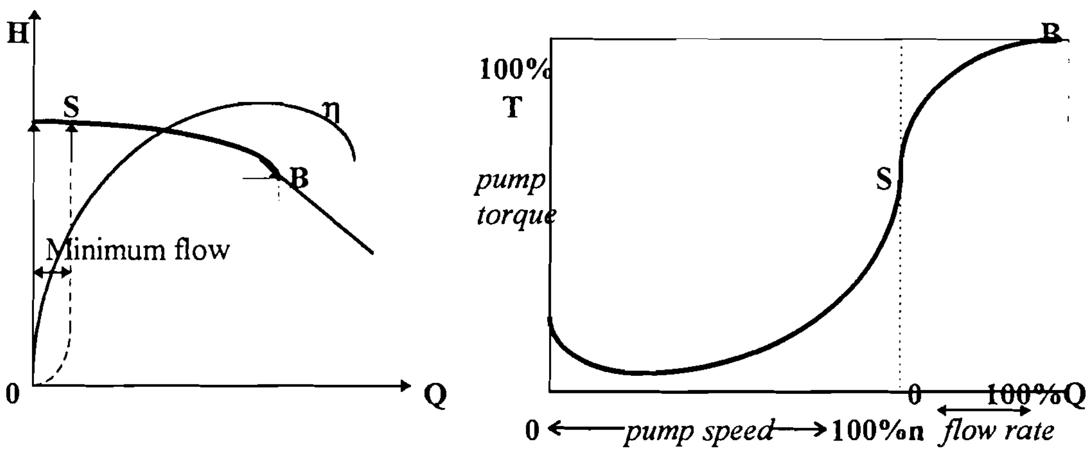


Figure 3.10: Starting against closed valve. B is duty point.

3.8.2 Start-up against closed non-return valve with the delivery valve open

When starting, the non-return valve opens at point *A* (see figure 3.11), when the counterpressure acting on it is reached. Delivery performance, i.e. flow rate and pressure, rises along the system head curve as a function of speed.

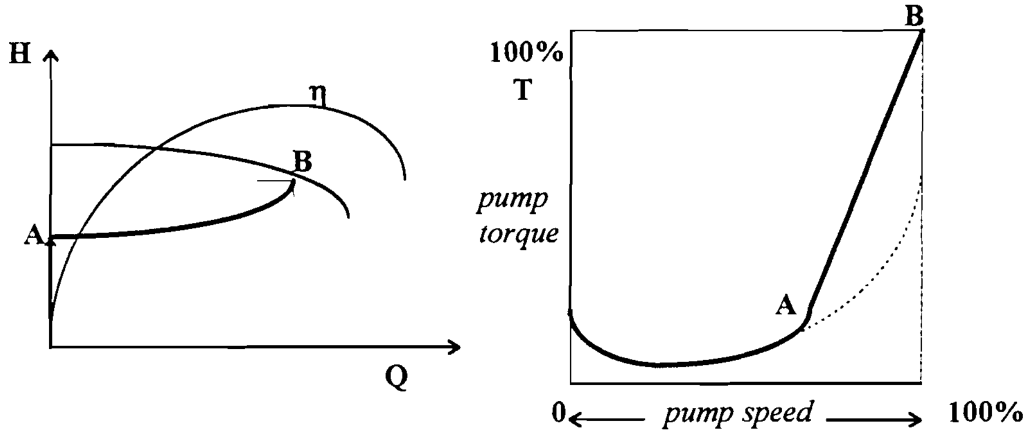


Figure 3.11: Start-up against closed non-return valve with delivery valve open.

3.8.3 Start-up with delivery valve open but no geodetic head

Here, the torque is a parabola through duty point *B* (see figure 3.12), provided that the pipe is short and that the time needed to accelerate the water column coincides with the pump start-up time

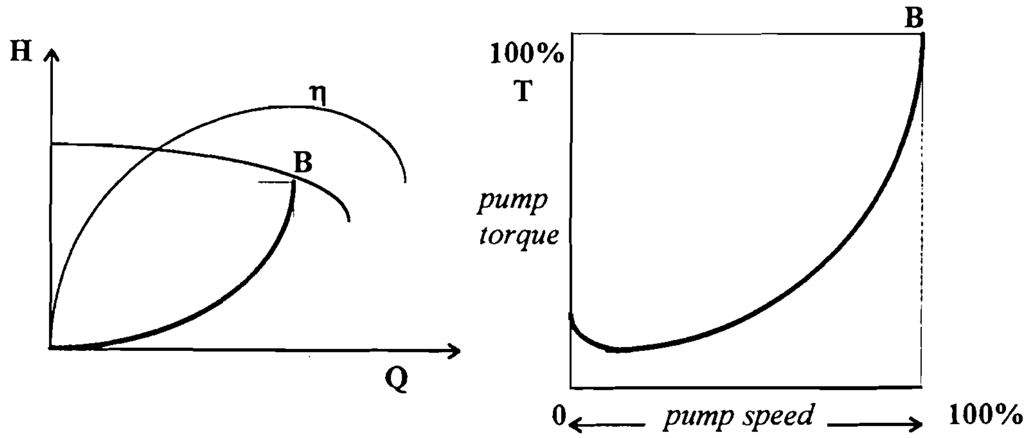


Figure 3.12: Start-up with delivery valve open but no geodetic head.

Chapter 4
Electric motors

4.1 General

The functioning of all electric machines is based on the physical laws of induction. The following paragraphs wil deal with the motor characteristics relevant to the operation of pumps.

4.2 Three-phase motors

When a centrifugal pump is driven, three different three-phase motors can be selected:
1. squirrel-cage motors;
2. slip-ring induction motors;
3. synchronous motors.

The centrifugal pumps are driven by a the first one, the squirrel-cage motor. So only information concerning this type of motor will be given below.

4.3 Applications for three-phase asynchronous squirrel-cage motors

In a asynchronous motor the rotor rotates asynchronously with the rotating stator field. The relative speed difference between the rotor and the rotating field of the stator (synchronous speed n_S) is referred to as *slip*. In table 4.1 some characteristic figures for slip of asynchronous motors operating at full load are given.

Table 4.1: Motor output vs. slip at full load.

Motor output (kW)	1	10	100	1000
Slip at full load (guide values)	5÷8%	2÷4%	1÷2%	0.8÷1%

Due to its robust construction, the squirrel-cage motor can be regarded as a general purpose type. It is suitable for a large number of working machines and is therefore utilized in all branches of industry.

In general, a high speed motor is cheaper for a given torque value. When a gear train is used, the cost of motor plus gear train must be taken into account when determining the appropriate motor speed.

A wide range of special versions facilitate the matching of motors to their applications still further. They include:
- stepped speed designs (pole change motor);
- specially matched torque characteristics;
- special types of construction and protection;
- infinitely variable speed facility obtained by electronic control via converter-controlled voltage and frequency regulation.

4.4 Boundary conditions affecting motor sizing

Selection of a motor is dependent on the power requirement, speed and drive conditions of the centrifugal pump. Motor size is also influenced by the type of current, the voltage and the frequency of the electricity supply grid. The following additional information is necessary to assure appropriate sizing:
- the requirements imposed by the centrifugal pump;
- the ambient conditions;
- the grid connection data;
- any special regulations in force.

These items will now be discussed in detail.

4.4.1 Requirements imposed by the centrifugal pump

a. Type of centrifugal pump.

b. Power requirement or indicator diagram (as a function of flow rate, speed or time).

c. Function;
Working cycle (switching frequency);
Gear.

d. Speed;
Speed adjustment range;
Direction of rotation.

e. Start-up with or without load;
Load moment curve;
Mass moment of inertia.

f. Control conditions;
Control magnitude;
Precision.

g. Any mechanical or electrical braking;
Braking torque curve.

h. Installation;
Fastening, foundation;
Physical configuration;
Force transfer to centrifugal pump;
Additional axial and radial forces incident on the end of the motor shaft;
Shaft end dimensions.

4.4.2 Ambient conditions

a. Ambient temperature;
Temperature of cooling medium (air, water);
Contamination of air - acid vapours, moisture, dust;
Analysis of water.

b. Special conditions;
Explosive gas mixtures;
Fire-damp (methane);
Installation in open air;
etc.

c. Altitude of installation (metres above sea level).

d. Type of protection.

e. Shaking.

f. Additional regulations;
Motor noise level;
Motor vibrations;
etc.

4.4.3 Grid connection details

a. Type of current, voltage, voltage fluctuations, frequency.
b. Safe operating values: startup current, power factor.

4.4.4 Special specifications

a. National regulations.
b. Climatic conditions.
c. Special instructions (operation, storage, transport, servicing etc.).

4.5 Starting characteristics of three-phase squirrel-cage motors

During running-up to operating speed, the torque of an asynchronous motor varies. Typical characteristic curves of a three-phase squirrel-cage motor is shown in figure 4.1.

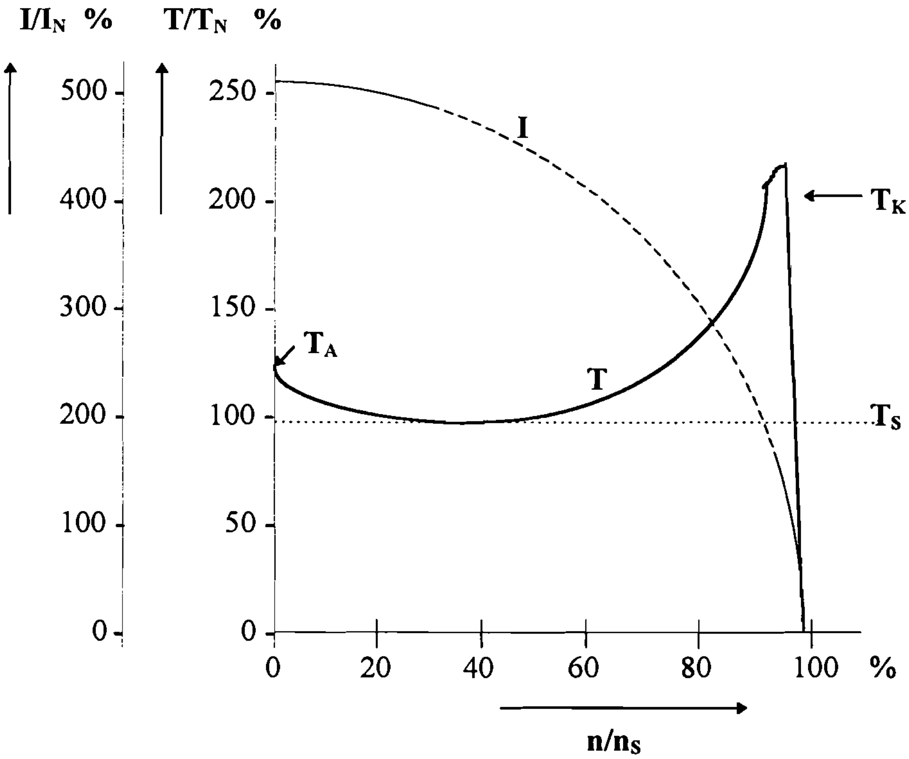


Figure 4.1: Typical starting characteristics of three-phase squirrel-cage motors.

In figure 4.1 the following abbreviations are used:

T:	Torque;	T_N:	Nominal torque;
n:	Speed;	n_S:	Synchronous speed;
I:	Starting current;	I_N:	Nominal current;
T_A:	Starting torque at standstill;		
T_S:	Pull-up torque as minimum value during running-up, where T_S is less than T_A;		
T_K:	Pull-out torque as maximum value during running-up.		

The characteristic curve in the case of squirrel-cage rotors is dependent on rotor slot shape; hence it can be matched to operating conditions by appropriate choise of these.

When directly connected at full grid voltage, three-phase squirrel-cage motors take up a transient current which can amount to between four and six times rated current, depending on power and the number of poles. During run-up the current decreases to a value corresponding to the load, e.g. to rated current when rated load is applied. The starting current can be reduced by special start-up procedures (when pump start-up torque so permits).

In the case of centrifugal pumps, the starting torque curve of the pump must be taken into account. This will be dealt with later.

4.6 Choice of starting method

When the supply grid characteristics and the drive permit direct connection, this starting procedure should be used exclusively. In other cases the most suitable starting methods available should be chosen. In table 4.2 the following methods are compared with direct connection (a):

(b): Starting via choke;
(c): Starting via starting transformer;
(d): Starting via transformer.

Table 4.2: Various starting methods compared with direct connection: [Sulzer, 1994]

Starting method	a	b	c	d
Working voltage U/U_N (across motor terminals)	1.0	0.8	0.8	0.8
Starting current I/I_N	4.8	3.84	3.07	3.84
Starting torque T_A/T_N	0.55	0.35	0.3	0.35

The most suitable starting method should be established in consultation with the pump, motor and electricity suppliers.

4.7 Speed control

The speed of three-phase squirrel-cage motors can be changed:

- in steps by *pole switching*;
- voltage change
  by phase control of the working voltage (only applicable for low ratings);
- continuously by *frequency changing*
  by means of converter or transformer (up to a few MW output), control range 1:5 and above (see figure 3.2)

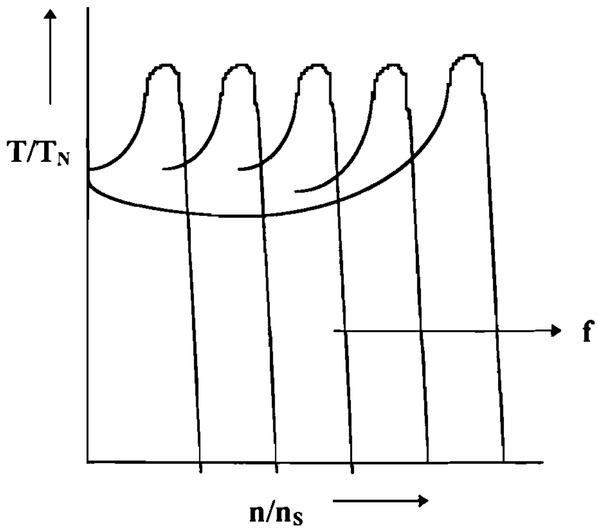


Figure 4.2: Speed control by change of frequency. f = frequency of working voltage U (voltage U to be changed proportionally to the frequency).

4.8 Power uptake

The active power absorbed (P_1) in kW and the reactive power (P_S) in kW are given in equation 4.8.1:

$$P_1 = \frac{P_N}{\eta_{mo}} \quad \text{[kW]} \qquad (4.8.1.a)$$

$$P_S = \frac{P_N}{\eta_{mo} \cdot \cos\varphi} \quad \text{[kVA]} \qquad (4.8.1.b)$$

where P_N = the nominal output (at motor shaft end) [kW]
η_{mo} = efficiency of the motor
$\cos\varphi$ = power factor

The nominal current (I_N) is calculated from:

$$I_N = \frac{1000 \cdot P_1}{U_N \cdot \sqrt{3} \cdot \cos\varphi} \quad [A] \qquad (4.8.2)$$

where U_N = nominal voltage

The power factor is a measure of the apparent power which is converted into a different form of energy (utilisation factor).

In general, it can be stated that efficiency and power factors [Sihi, 1980]
- Increase as the rated power of the motor increases;
- Decrease with decreasing load on the motor.

This should be kept in mind when selecting a motor for a pump drive since the inclusion of excessively high power margins in determining the pump power requirements, will mean that the motor will run permanently in the part-load range with poor efficiency and a low power factor.

The power factor is dicussed in detail in paragraph 10.4.

4.9 Type of working area

It is only possible to classify working areas into one of the following types. after more precise knowledge of the local and operating conditions is obtained.
- **Dry working areas** are rooms or places in which no condensation normally occurs, or in which the air is not saturated with moisture;
- **Damp and wet working areas** are rooms or places in which the safety of the machinery cab be adversely affected by moisture, condensation, chemical or other influences;
- **Fire-hazard work areas** are rooms or places within rooms or in the open, where there is a risk, as a result of the conditions of operation and space available, that readily inflammable materials in quantities constituting a danger may come sufficiently close to electrical plant to produce a fire risk;
- **Agricultural work areas** cover rooms and other places which are used for agricultural or similar purposes, and in which there is an increased risk of fire due to the presence of readily ignitable materials, or an increased risk of accident due to the affect of moisture, dust, chemically aggressive vapours, acids or salts on the insulation of the electrical plant;
- **Workshops subject to explosion hazards**. This is the working area for the electric motors which drive the pumps. So this area will be discussed in detail.

4.10 Explosion protection

4.10.1 General

In industrial production processes, in particular in the chemical industry and in refineries, gases, vapours, smoke and dusts are gererated, which in conjuction with asmospheric oxygen may form an explosive mixture. Under certain conditions such a mixture may ignite and explode.

The composition and concentration of mixture (ignition quality) in addition to the source of ignition (e.g. electrical discharges, overheated motors) play an important role. The results of such explosions are frequently substantial, material damage as well as loss of life often results. For this reason protective measures designed to prevent explosions are prescribed by law.

According to the 'Statutory regulations concerning electrical installations in locations subject to explosion hazards (ExVO)" electrical equipment may be used in locations subject to explosion hazards only if the following conditions are installed:

- They must be approved for the gases and vapours which are present;
- They must be subject to a detailed inspection by the manufacturer ensuring that they are in agreement with the type approval;
- They must bear the marks and particulars laid down by the relevant official authority.

For the classification of the requirements for explosion protection in the design of a motor-driven pump, it is insufficient to state only the discharge conditions and the physical properties of the liquid. Due to the local conditions, other sources of hazards have often to be taken into consideration for the necessary protective measures.

The decision as to whether a particular location in the open air or in a confined space is to be considered subject to explosion hazards, within the meaning of the relevant regulations and provisions, has to be made by the user and, in case of doubt, by the relevant supervising authority.

4.10.2 Classification in zones of potentially explosive atmospheres

Depending on the duration and frequency of the presence of a dangerous explosive atmosphere a distinction is made between the following zones of potentially explosive atmospheres:

- **Zone 0**: covers areas where a dangerous explosive atmosphere is continuosly present or present for long periods.
  The use of electric motors, irrespective of any protection type is forbidden. Exceptions can be made only by the local supervising authority;

- **Zone 1**: covers areas where a dangerous explosive atmosphere occurs in a range characterised as between 'occasionally' and 'frequently'.
  Motors of the 'flameproof enclosure' type (Ex)d must be used if an explosive atmosphere of potentially dangerous proportions occurs frequently. Where these conditions are an occasional occurrence motors of the 'increased safety' type (Ex)e can be used;
- **Zone 2**: covers areas where the dangerous explosive atmosphere occurs rarely, and if it occurs, it will only exist for a short period.
  Motors of 'flameproof enclosure' type (Ex)d and 'increased safety' type (Ex)e are permissible. In some cases motors without explosion protection may be used.

4.11 The pump in relation to the electric motor

4.11.1 Power, losses and efficiencies

In figure 4.3 the pump and the electric motor are connected. The different losses and powers are given.

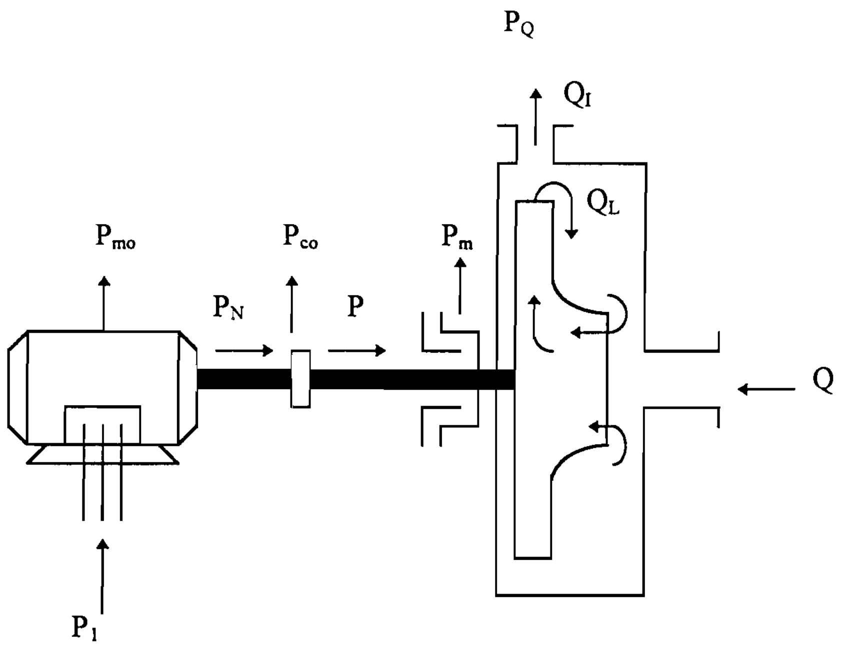


Figure 4.3: Power and losses between a pump and an electric motor.

In figure 4.3 the following abbreviations are used (see also paragraph 3.4 and 4.8):
Q_I = impeller flow; Q_L = leakage flow;
Q = useful flow rate;
P_{mo} = losses in the motor; P_{co} = losses in connection;
P_m = mechanical losses; P_N = the nominal output (at motor shaft end);
P_I = active power absorbed; P = power input (at the pump drive shaft).

By combining the different efficiencies from paragraph 3.4 and 4.8 the overall- or pumpefficiency is obtained:

$$\eta_{tot} = \eta_{mo} \cdot \eta_{co} \cdot \eta_m \cdot \eta_{hR} \cdot \eta_v \tag{4.11.1.1}$$

where η_{mo} = motor efficiency;
η_{co} = connection efficiency;
η_m = mechanical efficiency;
η_{hR} = combination of hydraulic and disc friction efficiency;
η_v = volumetric efficiency.

4.11.2 Start-up

During start-up the torque (T_p) demanded by the pump should be less than the motortorque (T_m). These characteristic torques are given in figure 4.4.

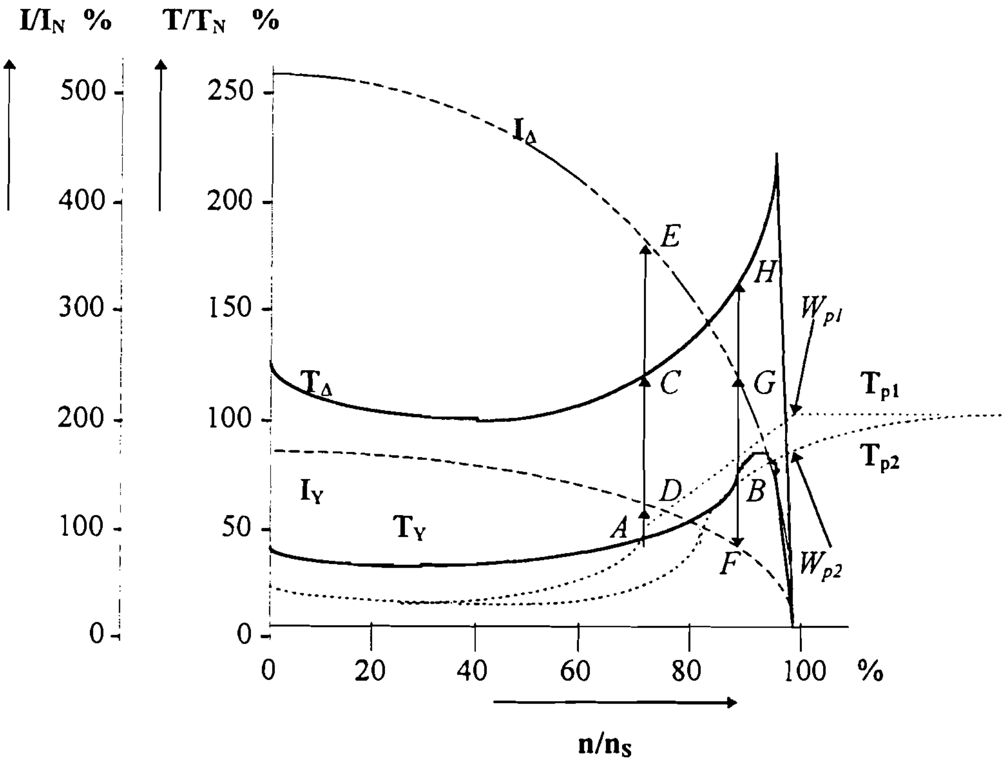


Figure 4.4: Characteristic curves for a centrifugal pump combined with an electric motor.

In figure 4.4 the following abbreviations are used:

- T_Δ = Motortorque by a deltaconnection;
- T_Y = Motortorque by a starconnection = $T_\Delta / 3$;
- I_Δ = Start-up current by a deltaconnection;
- I_Y = Start-up connection by a starconnection = $I_\Delta / 3$;
- T_{p1} = Torque demanded by the pump for a start-up with the delivery valve open;
- T_{p2} = Torque demanded by the pump for a start-up with a closed valve;
- A = Point where T_{p1} equals T_Y;
- B = Point where T_{p2} equals T_Y;
- W_{p1} = Duty point for a start-up with the delivery valve open;
- W_{p2} = Duty point for a start-up with a closed valve.

When the motor is started in a star-delta-connection, the motor starts with the lower T_Y and I_Y and switches to the higher T_Δ and I_Δ, when the torque demanded by the pump is too high for the T_Y.

For a start-up with the delivery valve open the pumptorque equals the motortorque in starconnection in A. As the motortorque should be bigger than the pumptorque, the motor shall switch from a star- to a deltaconnection ($A \rightarrow C$). As a result the start-up current will also change ($D \rightarrow E$). In point D the start-up current is ± 125% of I_N. In point E the start-up current is increased to 3 x 125% = **± 375%** of I_N.

In the case for a start-up against closed valve the motor will also switch from a star- to a deltaconnection ($B \rightarrow H$ and $F \rightarrow G$) only this time by a higher speed. In point F the start-up current is ± 80% of I_N. In point G the start-up current is increased to 3 x 80% = **± 240%** of I_N.

It is noticed that a start-up against a closed valve will cause a reduction in power absorbed.

Sometimes during start-up it happens at a specific speed n that the motortorque is less than the pumptorque. For example when suddenly the demanded flow rate a lot increases. At that moment the specific speed can't be increased and the starconnection must be switched to a deltaconnection. If this happens at a low specific speed a high start-up current will be present.

In the case (also at AFS) where centrifugal pumps are parallel connected, one should try to switch to an extra pump at the right moment to keep the start-up current as low as possible.

4.12 The different electric motors at Rijk

The Sulzer CZ 100-315 centrifugal pumps are driven by electric motors. These motors are not all the same. Table 4.3 shows the type of motor which is used by a pump at RijkI and RijkII (see also appendix A2 and A3 for the place of these pumps).

It is noticed that there are ten pumps which have a nominal output power P_N of 110kW (Q_{max} = 292 m^3/h) and the other ones 120kW (Q_{max} = 345 m^3/h) (see appendix C1 or figure 3.2).

Table 4.3: The different motortypes at Rijk I (P101-P112) and Rijk II (P201-P203) and some specifications.

Pump	Motortype	P_N (kW)	U_N (V)	I_N (A)
P101-P104	Hemaf 4K 315 SWT 20	110	380 S	195
P105-P106	Loher A 315 SA-2	110	380 S	198
P107-P110	Loher EMGV-315LB-0	110	380 S	197
P111-P112	Loher EMGV-315MD-	120	380 D	209
P201-P203	Loher DNSW-315MB-	120	415 D	215

Chapter 5

Flow control of pumps

5.1 Introduction

In principle, the flow rate of a pumping system can be controlled by shifting its duty point. This can be achieved in three ways:
1. By changing only the pumping system characteristic $H_S(Q)$, and keeping the same pump characteristic $H(Q)$;
2. By changing only the pump characteristic $H(Q)$, while keeping the same characteristic $H_S(Q)$ for the pumping system;
3. By simultaneously changing both characteristics $H_S(Q)$ and $H(Q)$.

When choosing the control procedure, several aspects should be kept in mind, particulary including economy, reliability and the automation potentialities. As a general rule, the control action of a pumping plant extends only over the flow capacity. The major index defining the quality of the flow rate control is denoted by ΔQ and defined as the ratio of the minimum capacity (Q_{min}) to the maximum one (Q_{max}) that can be achieved through the control procedure under consideration:

$$\Delta Q = \frac{Q_{\min}}{Q_{\max}} \tag{5.1.1}$$

Now, if the Q_{min} is taken as the basic quantity, equal to unity, the control range can be expressed as a ratio such as 1:10, 1:20, etc.

5.2 Control actions based on changes in the system characteristic

These control actions are all based on alteration of the pumping system characteristic ($H_S(Q)$), while keeping unchanged the pump characteristic ($H(Q)$). In the literarure they are called *quantitative controls*, because they are achieved through very simple means, but are associated with high energy losses. As a consequence this reduces the overall efficiency of the pumping plant.

Some control actions will now be discussed:
1. Control action by throttling of the fluid flow;
2. Control action through by-passing the fluid flow;
3. Control action through flow-off set.

5.2.1 Control action by throttling of the fluid flow

In this case the sytem head loss, and hence the dynamic head (H_{dyn}) is increased by means of a discharge control valve. The steepness of the system characteristic curve increases causing the intersection with the characteristic curve of the pump to occur at a lower flow rate.

Naturally, energy losses occur in the control valve. This means that continuous operation with throttling is uneconomic. The lowest throttling losses occur if the pump characteristic is flat. For this reason, control by throttling is mainly applied to radial flow pumps, bearing in mind that in their case the power absorbed also decreases with a reduced flowrate. Where control by throttling appears an attractive method of control with regard to the initial costs of the control system, investigations should be made into the economics of the method, since running costs particularly at high power inputs can be high.

In figure 5.1 the variation due to throttling is given.

5.2.2 Control action through by-passing the fluid flow

With bypass control a recirculation line is arranged parallel with the pump through which part of the flow is bled off from the delivery side of the pump and returned to the suction side (see fig. 5.2a). In accordance with the characteristic curve of the bypass, the system characteristic curve increases to a larger flowrate $Q_{tot} = Q_{bypass} + Q_A$ (see fig. 5.2 b). As a result the flowrate of the pump increases from Q_1 to Q_{tot} and the net flow delivered through the system decreases from Q_1 to Q_A.

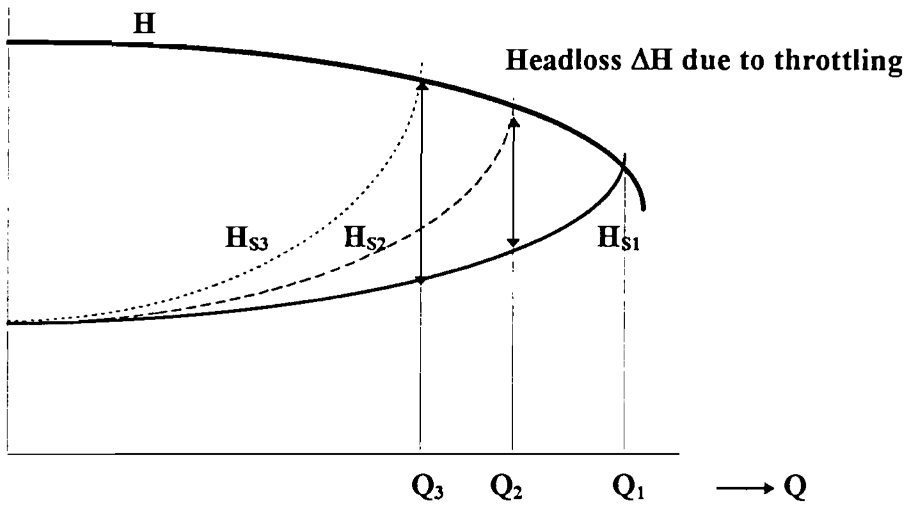


Figure 5.1: Variation of flowrate due to throttling.

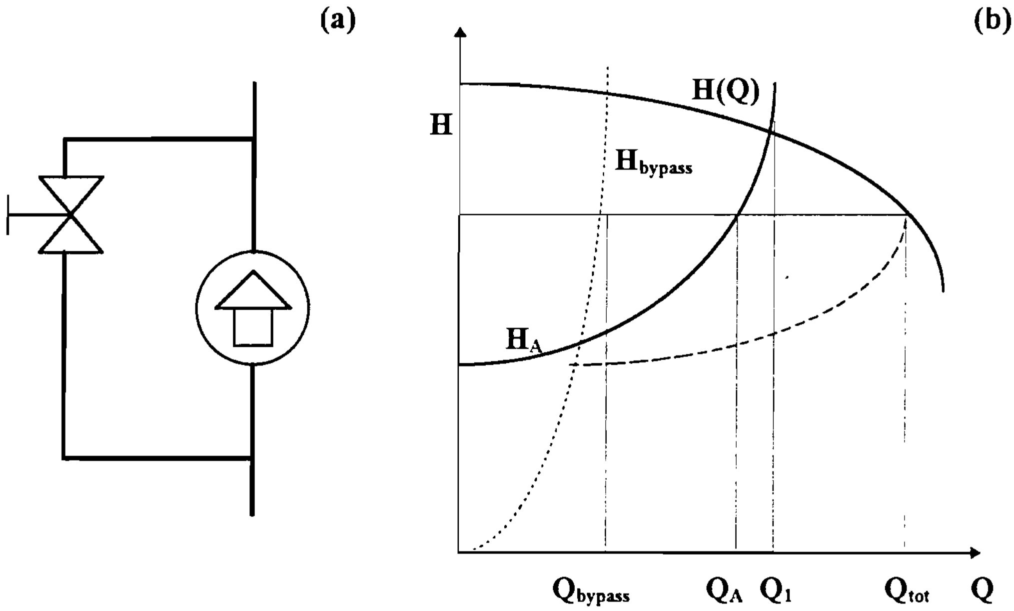


Figure 5.2: Flow control by means of a bypass;
(a) process flow diagram;
(b) diagrammatic representation.

In order to prevent the danger of excessive heating of the pumped liquid where large bypass flows are concerned, the liquid should be recirculated into the suction vessel and not into the suction pipe.

Reduction of the net flowrate by means of bypass is particularly recommended for side-channel and axial flow pumps, as the power absorbed by the pump decreases with increasing rate of flow.

5.2.3 Control action through flow-off set

This action is achieved by changing the static characteristic $H_{st}(Q)$ of the pumping system during starting and stopping periods. It requires an intermittent duty for the pump, the maximum recurrence of the on-off periods being limited by the compensating action of a hydraulic accumulator (hydro-pneumatic tank).

The hydraulic accumulator is branched on the discharge side. Its purpose is to off-set the difference between the flowrate required by the system and that pumped by the pump as the hydraulic accumulator is filled up or drained off, the static head of the pumping system is accordingly increased or lowered.

With such control actions, the recommended pumps are those with a steep characteristic H(Q), because in this case the flow rate control range ΔQ is rather narrow, and hence the pumps will be close to their maximum efficiency. If pumps with a flat characteristic are used, the resulting flow rate control range will be broad and the corresponding efficiency values, found on the $\eta(Q)$ curve, will be low as they represent extreme working points on the characteristic.

The off-set control offers some benefits (in the first place, low operating costs) but requires at the same time high investment costs (due to the exaggerated volume of the hydraulic accumulator). Since the volume of the latter is proportional to the unit output of the controlled pump, this method is recommended only for pumping systems fitted with low flow rate pumps ($Q \approx 36$ m^3/h).

5.3 Control actions based on changes in the pump characteristic

These contol actions are carried out by changing the pump characteristic, while keeping the same characteristic for the pumping system. In practice, they are achieved by means of electric adjustable speed drives, or special mechanical gears such as:

- by altering the position of impeller blades;
- by adjustable inlet guide vanes.

Whatever the control solution chosen, the energy losses are small and the overall efficiency of the pumping plants is considerably increased over those of unadjustable pumps (by 30 per cent); likewise, the electric power requirements are substantially reduced **(by up to 25 per cent)** [Ionel, 1986].

5.3.1 Speed variation

The rotational speed of the pump can be varied with the help of adjustable-speed drives. If the driving speed of the pump is varied, its characteristic H(Q) is changed, and the flow discharged by the pump is changed in direct proportion. This method has the best results when it is applied by pumps having the concave side of the characteristic H(Q) upwards, such as centrifugal pumps.

Such characteristics ideally overlap on the pumping system characteristic and result in considerably higher efficiencies for the pumping plant.

Three different cases for a pumping system can be distinguished:
1. *System having a static head only*. This is a relatively seldom met case in practice. However, one is often confronted with systems whose maximum pumping head is the static head. In order to control the flow rate, the pump in such cases has available only a narrow speed control range ($\Delta n = n_{min} / n_{max} \leq 1 : 1.25$), and within the minimum capacity range the pumping plant develops only a low efficiency, because the resultant artificial characteristic H(Q) of the pump lies far enough from the characteristic of optimal efficiency.
2. *Systems having a dynamic head only*. This typical system applies in particular to recycling pumping plants. Indeed, to control the flow rate in such a case, the pump should cover a wide speed control range ($\Delta n < 1:2$). On the other hand, the pumping plant performs at a high efficiency over the range of minimum capacities because the resultant artificial characteristic is perfectly coincident with the optimal efficiency characteristic. As a consequence ΔQ is theoretically unlimited. These operating conditions are ideal for the control of pumping plants fitted with variable-speed pumps, in contrast to those plants fitted with pumps driven at a constant speed, for which such conditions are totally inadequate.
3. *System with both a static and a dynamic head*. This is the case generally met in practical pumping plants. To control the pumping plant capacity in such a case, the variable-speed pump should cover a medium-width speed control range, while performing at acceptable efficiencies over the range of minimum flow rates; this is because the artificial characteristic obtained by pump control is close to that of the optimal efficiency. As a consequence ΔQ is but little restricted.

5.3.2 Altering the position of impeller blades

This procedure is characteristic of axial-flow pumps with variable-pitch blades and is based on the alteration of performance characteristic curves through changing the impeller blades angle of tilt which, in turn, brings about a change in the profile hydraulic characteristics. This procedure is beset with difficulties in practice and is also very expensive; it is therefore seldom applied.

5.3.3 Comparison of the different control actions

Compared with throttling or by-passing, the rotational speed control has no decrimental effect on the efficiency. Indeed, if the rotational speed of a pump is changed, say, to 90% n_r, its output drops to 90% Q_r, its pumping head to 81% H_r, and the power to 73% P_r, but the efficiency keeps pace with the rated efficiency (η = 100% η_r), as compared with for Q = 90% Q_r (and H = 107% H_r) for throttling, and with η = 93% η_r for 115% Q_r (and H = 107% H_r) for by-passing. Thus there are considerably economic benefits of the control actions based on speed variation.

At the AFS pumping plant, Sulzer CZ 100-315 pumps are used. These pumps are radial-flow pumps and have a specific speed of 21 (see also paragraph 3.3.2). The following control actions are **not** recommended:

- By-passing;
- Flow off-set;
- Altering the position of the impeller blades.

By using the rotational speed method, it is important to know which shape the pumping system characteristic has. So, in the next chapter this characteristic will be calculated.

Chapter 6

Pumping systems

6.1 General

A pumping system is meant to transport liquid from a lower energy level to a higher one. Transport is achieved with the help of two distinct lines: the suction line and the discharge line. The energy required for the liquid transfer from the lower to the higher energitical head is provided by pumps. This energy is represented by the hydrodynamic load H_t and is called *the total pumping head of the system*.

H_t comprises in general the following (see figure 6.1):

- Static head: $H_{geo} = z_a - z_e$;
- Difference between pressure head: $(p_a - p_e)/(\rho \cdot g)$;
- Difference between velocity head: $(v_a^2 - v_e^2) / 2g$;
- Head loss, which is split into:
  - suction or inlet pipe losses H_{vs} of the system, from the system inlet A_e to the pump inlet A_s, including possible inlet losses and losses due to valves and fittings, etc.
  - loss of head H_{vd} in the delivery pipework of the system, from the pump outlet A_d to the system outlet A_a, including possible inlet losses and losses due to valves and fittings, etc.

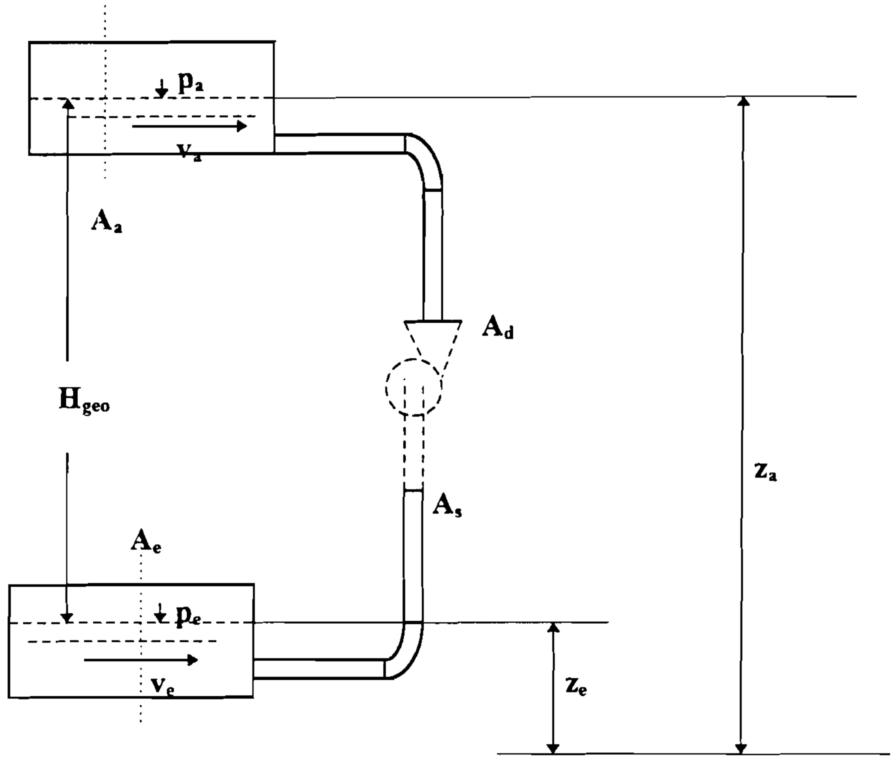


Figure 6.1: The total piping system.

This leads to

$$H_t = (z_a - z_e) + \frac{p_a - p_e}{\rho \cdot g} + \frac{v_a^2 - v_e^2}{2 \cdot g} + H_v \tag{6.1.1}$$

where: - the first two parts the static component represent and are independent of the flowrate;
- the last two parts the dynamic component represent and are dependent of the flowrate.
- $v_a = Q / A_a$ [m/s];
- $v_e = Q / A_e$ [m/s];
- p_a, p_e [N/m^2];
- ρ [kg/m^3];
- Q [m^3/s].

It is expedient to select, for inlet and outlet sections of the system, reference points for which the elevation, pressure and cross-sectional area are known. Normally this applies to the suction chamber or container on the inlet side of the pump or relevant point of discharge in the delivery pipeline.

For the situations at Rijk and at Oost the total head loss (in the pipes and the valves) at a certain flow will be calculated.

6.2 Pipes

6.2.1 Head loss H_v in straight pipes

For pipes of circular cross-section which are completely filled, the head loss H_v can be calculated according to *Darcy-Weisbach* as follows:

$$H_v = \lambda \cdot \frac{l}{D} \cdot \frac{v^2}{2 \cdot g} \cdot 10^3 \qquad [\mathrm{m}] \qquad (6.2.1.1)$$

where λ = Coefficient of friction;
l = Pipe length [m];
D = Diameter of the pipe [mm];
v = Mean velocity = flow rate / cross-section area of pipe [m/s]

The coefficient of friction λ is a function of the non-dimensional Reynolds number

$$\mathrm{Re} = \frac{D \cdot v}{\nu} \cdot 10^3 \qquad \text{or} \qquad \mathrm{Re} = \frac{354 \cdot Q}{D \cdot \nu} \cdot 10^3 \qquad (6.2.1.2)$$

where ν = kinematic viscosity [mm^2/s];
Q = flowrate [m^3/h].

For Re < 2320, i.e. for laminar flow, the law of *Hagen-Poiseuille* applies, independent of the roughness of the pipe bore.

$$\lambda = \frac{64}{\mathrm{Re}} \qquad (6.2.1.3)$$

For Re > 2320, i.e. for turbulent flow, a condition which is present in nearly all normal cases, the value λ amounts to:
(a)

$$\frac{1}{\sqrt{\lambda}} = 2 \cdot \log\left(\frac{\mathrm{Re} \cdot \sqrt{\lambda}}{2{,}51}\right) \qquad (6.2.1.4)$$

for the limiting case of hydraulically smooth pipes, where λ only depends on Re and where its value can be calculated by the above equation.

(b)

$$\frac{1}{\sqrt{\lambda}} = 1{,}14 - 2{,}0 \cdot \log\frac{k}{D} \qquad (6.2.1.5)$$

where k is the internal surface roughness of the pipe.

For the limiting case of hydraulically rough conditions, where λ only depends on the internal roughness of the pipe and the diameter of the pipe, its value can be calculated using the above equation.

For the usual pipe materials, diameters of pipes, and flow velocities, a condition between *hydraulically smooth* and *hydraulically rough* exists. For this transition range the coefficient of friction λ follows the relation below, accredited to *Prandtl-Colebrook*:

$$\frac{1}{\sqrt{\lambda}} = -2 \cdot \log\left(\frac{2{,}51}{\mathrm{Re} \cdot \sqrt{\lambda}} + \frac{k}{D} \cdot \frac{1}{3{,}71}\right) \tag{6.2.1.6}$$

In this range the Reynolds number Re and the condition of the pipe bore, expressed by the absolute roughness k or the relative roughness k/D, respectively, will affect the magnitude of the coefficient of friction λ.

6.2.2 Head loss H_v for turbulent flow, for liquids with a $\nu \neq 1.236$ mm^2/s

To determine λ from equation 6.2.1.6 is difficult and time-consuming. It is therefore more expedient to determine the head loss H_v with the assistance of tables 1 and 2 (see appendix D) which are based on this equation.

Table D.2 was established using the following values:

- an absolute roughness k = 0.1 mm for the internal surface of the pipe (new cast iron pipe, bitumen coat on the inside);
- a kinematic viscosity = 1.236 mm^2/s (viscosity of pure water at 12°C). The values obtained for water are sufficiently accurate to be used for water and other liquids of similar viscosity at normal ambient temperature;
- length of the pipe l = 100m.

The correction of the head loss H_v for turbulent flows, for liquids with a kinematic viscosity $\nu \neq 1{,}236$ mm^2/s is carried out in three steps (a) to (c):

(a)

$$Q = Q_x \cdot \left(\frac{1{,}236}{\nu}\right) \tag{6.2.2.1}$$

where subscipt x denotes the values that apply at the given kinematic viscosity.

(b)
If the absolute roughness of the actual pipe is different from 0.1 mm (see table 6.1), then the head loss obtained from table D.2 for k = 0.1mm has to be multiplied with a correcting factor as follows:

$$H_v = H_{v(k=0.1)} \cdot f \qquad (6.2.2.2)$$

to be established from table D.1
to be established from table D.2

(c)
H_{vx} can now be calculated from:

$$H_{vx} = H_v \cdot \left(\frac{v}{1{,}236}\right)^2 \qquad (6.2.2.3)$$

Table 6.1: Roughness values k for various pipe materials and various pipe conditions.

Material and type of pipe	Condition of the pipe	k in mm
new cast iron pipes	with bitumen lining without bitumen lining with cement lining	0.1 to 0.15 0.15 to 0.25 0.025
used cast iron pipes	uniform corrosion pits slight to heavy incrustations after several years operation cleaned	1 to 1.5 1.5 to 3
new seamless steel pipes	rolled or drawn	0.02 to 0.05
new longitudinally welded steel pipes		0.04 to 0.1
new steel pipes, lined	electroplated with zinc bitumen lined cement lined galvanized	0.1 to 0.15 0.05 0.025 0.01
used steel pipes	uniform corrosion pits slight incrustations mediun incrustations heavy incrustations	0.15 0.15 to 0.4 1.5 2 to 4
asbestos cement pipes	new, commercial grade	0.03 to 0.1
new concrete pipes	commercial grade, rendered commercial grade, medium commercial grade, rough	0.3 to 0.8 1 to 2 2 to 3
used concrete pipes	after several years operation with water	0.2 to 0.3
stoneware pipes, spun concrete pipes		0.25
drawn and pressed pipes of copper, brass, aluminium, glass plastics	new used	up to 0.01 up to 0.03

6.3 AFS piping system

6.3.1 General

First the head loss in the 32 inch pipe (D = 812,8 mm) will be calculated. The flow medium is kerosine ($\nu = 5\ mm^2/s$). The length of the 32 inch pipe in total is 3700m and it is made from steel (k = 0,05).

By using equation 6.2.1.2 the Reynolds number can be calculated:

$$Re = 87.11 \cdot Q \tag{6.3.1.1}$$

For $Q > 27\ m^3/h$ turbulent flow will arise.
Step (a) gives:

$$Q = Q_x \cdot 1{,}236/5 = 0.2472 \cdot Q_x \tag{6.3.1.2}$$

Step (c) gives:

$$H_{vx} = H_v \cdot 16.36 \cdot 1 / 100 \tag{6.3.1.3}$$

As table D2 is limited, one should use table D3 for the smaller flows. This table gives the unit head loss, in meters per one linear meter of pipe as function of diameter and the mean velocity.

6.3.2 Deriving the head-loss for the different pipes

In Appendix E1 the piping system from RijkI to the hydrant system is given. In this model the valves are also given. There are five different sizes of pipes used at AFS (8", 10", 12", 18", 20", 32"). The head loss of this pipes can be calculated by using the tables in appendix D. If one only wants to calculate **some** values for the head loss, this is a good and a approximate way. If one wants to calculate the head loss for a lot of values (Q), one should need a relation for the total head loss. As it is important for the situation at AFS to calculate the head loss for a lot of different flowrates, some relations for the different pipes will now be obtained. The following relations will be used:

$$\log H_v = a \cdot \log Q - b \tag{6.3.2.1}$$

$$\log(A \cdot B) = \log A + \log B \tag{6.3.2.2}$$

By using Table D2 the following relations for the different pipes are obtained:

The 32 inch pipe:

$$\log H_{v(k=0.1)} = 1.888 \cdot \log(0.2472 \cdot Q_x) - 7.244 \qquad (6.3.2.3)$$

The 20 inch pipe:

$$\log H_{v(k=0.1)} = 1.888 \cdot \log(0.2472 \cdot Q_x) - 6.153 \qquad (6.3.2.4)$$

The 18 inch pipe:

$$\log H_{v(k=0.1)} = 1.888 \cdot \log(0.2472 \cdot Q_x) - 6.01 \qquad (6.3.2.5)$$

The 12 inch pipe:

$$\log H_{v(k=0.1)} = 1.888 \cdot \log(0.2472 \cdot Q_x) - 5.07 \qquad (6.3.2.6)$$

The 10 inch pipe:

$$\log H_{v(k=0.1)} = 1.888 \cdot \log(0.2472 \cdot Q_x) - 4.7 \qquad (6.3.2.7)$$

The 8 inch pipe:

$$\log H_{v(k=0.1)} = 1.888 \cdot \log(0.2472 \cdot Q_x) - 4.139 \qquad (6.3.2.8)$$

As $k = 0{,}05$ step (b) is calculated as follows:

$$H_v = H_{v(k=0.1)} \cdot 0.93 \qquad (6.3.2.9)$$

Subsituting (6.3.2.9) in (6.3.1.3):

$$H_{vx} = H_{v(k=0.1)} \cdot 0{,}152 \cdot l \qquad (6.3.2.10)$$

where l is the length of the pipe.

6.4 Valves

6.4.1 Introduction

Two different kinds of valves are used in the piping system at AFS:
1. TK ball valves;
2. Twin Seal *double block and bleed* valve.

Double block and bleed is a means of establishing positive isolation between two sections of a pipe system using two stop valves and a bleed valve mounted between them. Closing the main valves and opening the bleed increased the security across the system. If the first valve failed, any leakage was diverted through the open bleed.

This performance level eliminates the (often hidden) costs of leaking valves, and is ideal for critical applications in manifold design, metering systems, tank farm isolation services etc.

For a valve the flow (liquid) coefficient equation is

$$Q = C_v \cdot \sqrt{\frac{\Delta p}{(s.g.)}} \tag{6.4.1.1a}$$

or

$$\Delta p = 0{,}8 \cdot \left(\frac{Q}{C_v}\right)^2 \tag{6.4.1.1b}$$

where	Q	=	flow [gallons per minute]
	Δp	=	pressure drop [p.s.i.]
	s.g.	=	specific gravity (kerosine = 0,8)
	C_v	=	flow coefficient, the volume of water in gallons per minute at 60°F that will flow through a given element with a pressure drop of one p.s.i.

By using the following relations

$$Q_{m3/h} = 0.227 \cdot Q_{G/m} \tag{6.4.1.2}$$

$$\Delta p_{meter} = 12{,}3 \cdot \Delta p_{bar} = 12{,}3 \cdot 0.069 \cdot \Delta p_{psi} = 0.8487 \cdot \Delta p_{psi} \tag{6.4.1.3}$$

an expression for the pressure drop [m] is obtained:

$$\Delta p = 13{,}18 \cdot \left(\frac{Q}{C_v}\right)^2 \tag{6.4.1.4}$$

In table 6.2 the flow coeffients for TK ball valves and Twin seal valves are given. In this table only the values that will be used are given (ANSI 150, reduced bore).

By combining flow coefficients the following relations can be used:

1. Flow in parallel:
$C_v = C_{v1} + C_{v2} + \ldots$

2. Flow in series:
$(1/C_v)^2 = (1/C_{v1})^2 + (1/C_{v2})^2 + \ldots$

Table 6.2: Flow coefficient values for TK ball - and Twin seal valves

Size of the pipe (inch)	C_v TK ball valves	C_v Twin Seal valve
8 x 6	2022	
10 x 8	4183	3500
12 x 10	7342	4000
18 x 16	20841	10900
20 x 18	28064	15700
30 x 26	51377	

Now the total losses in the piping systems
1. Tank at RijkIB or RijkIIB to the hydrant system;
2. Oost to Rijk.
will be discussed and an approximation of the pressure drop as a function of the flow will be calculated.

6.5 Piping system tank Rijk to hydrant system

A simpified scheme of the AFS fuel piping system from Rijk to Schiphol Centre is given in Appendix E1.

For the calculation of the minimum pressure drop the shortest way has been chosen.

6.5.1 Pump department RijkI to valve chamber P9a

A. Discharge pipe (12") of the pumps:

- Length 10 meter

$$H_v = 9{,}95 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(Qpomp))} \qquad (6.5.1.1)$$

- TK ball valve 12x10

$Q_{pomp} = Q_{tot}/(\text{amount of working pumps (m)}) \leq 288\ m^3/h$

$C_{v12x10} = 7342$

$$\Delta p_{meter} = 2{,}45 \cdot 10^{-7} \cdot (Q_{pomp})^2 \qquad (6.5.1.2)$$

B. 32'' pipeline to valve chamber P9a:

- Length: 1600 meter

$$H_v = 9{,}92 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(Qtot))} \tag{6.5.1.3}$$

- TK ball valve 30x26

$C_{v30x26} = 51377$

$$\Delta p_{meter} = 5 \cdot 10^{-9} \cdot (Q_{tot})^2 \tag{6.5.1.4}$$

6.5.2 Valve chamber P9a to different locations:

A. D-pier

- Total length 12" pipes: 300 meter

$$H_v = 2{,}77 \cdot 10^{-5} \cdot 10^{(1{,}888 \cdot \log(QD))} \tag{6.5.2.1}$$

- 3 TK ball valves (12x10) in series: $C_{vtot} = 4239$

$$\Delta p_{meter} = 7{,}33 \cdot 10^{-7} \cdot (Q_D)^2 \tag{6.5.2.2}$$

B. Centrum

- Total length 12" pipes: 1375 meter

$$H_v = 1{,}27 \cdot 10^{-4} \cdot 10^{(1{,}888 \cdot \log(Qcentrum))} \tag{6.5.2.3}$$

- 2 TK ball valves (12x10) in series: $C_{vtot1} = 5192$
- 3 Twin Seals (12 x10) in series: $C_{vtot2} = 2309$
- Total C_v: 2110

$$\Delta p_{meter} = 2{,}96 \cdot 10^{-6} \cdot (Q_{centrum})^2 \tag{6.5.2.4}$$

C. Valve chamber P16

- Length 32": 1150 meter

$$H_v = 7{,}14 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QE + QF + QG))} \tag{6.5.2.5}$$

- for E-pier (odd numbers), F-pier and G-pier:
  1 TK ball valve (30x26): $C_{v30x26} = 51377$

$$\Delta p_{meter} = 5 \cdot 10^{-9} \cdot (Q_E + Q_F + Q_G)^2 \tag{6.5.2.6}$$

6.5.3 Valve chamber P16 to different locations:

A. E-pier

- Length 18": 500 meter

$$H_v = 5{,}32 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QE))} \quad (6.5.3.1)$$

- 1 TK ball valve (18x16): $C_v = 20841$

$$\Delta p_{meter} = 3{,}03 \cdot 10^{-8} \cdot (Q_E)^2 \quad (6.5.3.2)$$

B. Valve chamber P18

- Length 32": 550 meter

$$H_v = 3{,}41 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QF - QG))} \quad (6.5.3.3)$$

6.5.4 Valve chamber P18 to different locations

A. F-pier

- Length 20": 150 meter

$$H_v = 1{,}11 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QF))} \quad (6.5.4.1a)$$

- Length 18": 150 meter

$$H_v = 1{,}6 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QF))} \quad (6.5.4.1b)$$

Total: $$H_v = 2{,}71 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QF))} \quad (6.5.4.2)$$

- 1 TK ball valve (20x18): $C_v = 28064$

$$\Delta p_{meter} = 1{,}67 \cdot 10^{-8} \cdot (Q_F)^2 \quad (6.5.4.3)$$

B. Valve chamber R9

- Length 32": 400 meter

$$H_v = 1{,}51 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QG))} \tag{6.5.4.4}$$

- To G-pier (odd):
  1 TK ball valve (30x26): C_{v30x26} = 51377

$$\Delta p_{meter} = 5 \cdot 10^{-9} \cdot (Q_G)^2 \tag{6.5.4.5}$$

6.5.5 Valve chamber R9 to G-pier

- Length 20": 250 meter

$$H_v = 1{,}91 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QG))} \tag{6.5.5.1}$$

- 1 TK ball valve (20x18): C_v = 28064

$$\Delta p_{meter} = 1{,}67 \cdot 10^{-8} \cdot (Q_G)^2 \tag{6.5.5.2}$$

6.5.6 Pumpdepartment RijkI via valve chamber P5 to Centrum

- Only possible via pump 101 and pump 102;
- Length 10" pipeline: 1220 meter

$$H_v = 2{,}62 \cdot 10^{-4} \cdot 10^{(1{,}888 \cdot \log(Qcentrum))} \tag{6.5.6.1}$$

- 1 TK ball valve (10x8): C_v = 4183
- 1 Twin Seal (10x8): C_v = 3500
  C_{vtot}: 2684

$$\Delta p_{meter} = 1{,}83 \cdot 10^{-6} \cdot (Q_{centrum})^2 \tag{6.5.6.2}$$

- Length 12" pipeline: 950 meter

$$H_v = 8{,}78 \cdot 10^{-5} \cdot 10^{(1{,}888 \cdot \log(Qcentrum))} \tag{6.5.6.3}$$

total losses in the pipelines:

$$H_v = 3{,}5 \cdot 10^{-4} \cdot 10^{(1{,}888 \cdot \log(Qcentrum))} \tag{6.5.6.4}$$

- 1 TK- valve (12x10): $C_v = 7342$
- 2 Twin Seals (12x10): $C_v = 2828$
  C_{vtot}: 2639

$$\Delta p_{meter} = 1{,}89 \cdot 10^{-6} \cdot (Q_{centrum})^2 \tag{6.5.6.5}$$

From these calculations it is noticed that the losses when pumping via valve chamber P5 are higher than when pumping via valve chamber P9a.

6.5.7 Tank Rijk IB / Tank Rijk IIB to pump department Rijk

A. Tank RijkIB to pumpdepartment

- length 20" suction line: 50 meter

$$H_v = 3{,}7 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QRijk2B))} \tag{6.5.7.1}$$

- 1 Twin Seal (20x18): $C_v = 15700$

$$\Delta p_{meter} = 5{,}35 \cdot 10^{-8} \cdot (Q_{Rijk1B})^2 \tag{6.5.7.2}$$

Q_{Rijk1B} (flow of tank RijkIB) + Q_{Rijk2B} (flow of tank Rijk IIB) = Q_{tot} (flow to hydrant system)

When m > 6 (more than six pumps are working) two tanks will be opened to make sure that enough fuel is brought to the hydrant system.

B. Tank RijkIIB to pump department

- length 20" pipeline: 100 meter

$$H_v = 7{,}4 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QRijk2B))} \tag{6.5.7.3}$$

- 1 TK ball valve (20x18): $C_v = 28064$
- 2 Twin Seal (20x18): $C_v = 15700$
  $C_{vtot} = 13702$

$$\Delta p_{meter} = 7{,}02 \cdot 10^{-8} \cdot (Q_{Rijk2B})^2 \tag{6.5.7.4}$$

6.5.8 Total losses tank RijkIB/RijkIIB to hydrant system

The relations from the paragraphs 6.5.1 to 6.5.6 will now be combined.

A. ***Losses in the pipes:***

$$
\begin{aligned}
H_{vtot} = {} & 3{,}7 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QRijk1B))} + 7{,}4 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QRijk2B))} + \\
& + 9{,}95 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(Qpomp))} + 9{,}92 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(Qtot))} + \\
& + 2{,}77 \cdot 10^{-5} \cdot 10^{(1{,}888 \cdot \log(QD))} + 1{,}27 \cdot 10^{-4} \cdot 10^{(1{,}888 \cdot \log(Qcentrum))} + \\
& + 7{,}14 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QE + QF + QG))} + 5{,}32 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QE))} + \\
& + 3{,}41 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(QF + QG))} + 2{,}71 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QF))} + \\
& + 3{,}42 \cdot 10^{-6} \cdot 10^{(1{,}888 \cdot \log(QG)} \qquad (6.5.8.1)
\end{aligned}
$$

B. ***Losses in Valves:***

$$
\begin{aligned}
\Delta p_{tot}\,[m] = {} & 5{,}35 \cdot 10^{-7} \cdot (Q_{Rijk1B})^2 + 7{,}02 \cdot 10^{-8} \cdot (Q_{Rijk2B})^2 + \\
& + 2{,}45 \cdot 10^{-7} \cdot (Q_{pomp})^2 + 5 \cdot 10^{-9} \cdot (Q_{tot})^2 + 7{,}33 \cdot 10^{-7} \cdot (Q_D)^2 + \\
& + 2{,}96 \cdot 10^{-6} \cdot (Q_{centrum})^2 + 5 \cdot 10^{-9} \cdot (Q_E + Q_F + Q_G)^2 + 3{,}03 \cdot 10^{-8} \cdot (Q_E)^2 + \\
& + 1{,}67 \cdot 10^{-8} \cdot (Q_F)^2 + 2{,}17 \cdot 10^{-8} \cdot (Q_G)^2 \qquad (6.5.8.2)
\end{aligned}
$$

where Q_D = Flow demanded at the D-pier;
Q_E = Flow demanded at the E-pier;
Q_F = Flow demanded at the F-pier;
Q_G = Flow demanded at the G-pier;
$Q_{centrum}$ = Flow demanded at centrum.

6.5.9 Calculation and checking of the head losses

In this paragraph the losses for different flows will be calculated to show how high the demanded system head is. In the calculations the following conditions are satisfied:

- From real measurements the following percent rates of flow at the different piers are obtained:
  D-pier: 5,4 %
  E-pier: 17,4 %
  F-pier: 33,3 %
  G-pier: 26,2 %
  Centrum: 17,7 %
- The height of the tank is not taken into account.

Table 6.3: The calculated head loss for different flowrates at the piers.

Q_{tot} [m^3/h]	Q_D [m^3/h]	Q_E [m^3/h]	Q_F [m^3/h]	Q_G [m^3/h]	$Q_{centrum}$ [m^3/h]	H_{Rijk1B} [m]	H_{Rijk2B} [m]
50	2,7	8,7	16,65	13,1	8,85	0,015	0,015
100	5,4	17,4	33,3	26,2	19,65	0,05	0,06
150	8,1	26,1	49,95	39,3	26,55	0,12	0,12
200	10,8	34,8	66,6	52,4	35,4	0,2	0,21
250	13,5	43,5	83,25	65,5	44,25	0,31	0,32
300	16,2	52,2	99,9	78,6	53,1	0,39	0,40
400	21,6	69,6	133,2	104,8	70,8	0,66	0,7
500	27	87	166,5	131	88,5	1,01	1,06
600	32,4	104,4	199,8	157,2	106,2	1,39	1,46
750	40,5	130,5	249,75	196,5	132,75	2,12	2,23
900	48,6	156,6	299,7	235,8	159,3	2,97	3,12
1100	59,4	191,4	366,4	288,2	194,7	4,33	4,56
1250	67,5	217,5	416,25	327,5	221,25	5,49	5,78
1375	74,25	239,25	457,88	360,25	243,38	6,57	6,92
1500	81	261	499,5	393	265,5	7,73	8,13
1650	89,1	287,1	549,45	432,3	292,05	9,25	9,74
1900	102,6	330,6	632,7	497,8	336,3	12,06	12,69
2150	116,1	374,1	715,95	563,3	380,55	15,22	16,02
2500	135	435	832,5	655	442,5	20,22	21,29
3000	162	522	999	786	531	28,50	30,01

The $H_{systemRijk2B}(Q)$ is also given in figure 6.2.

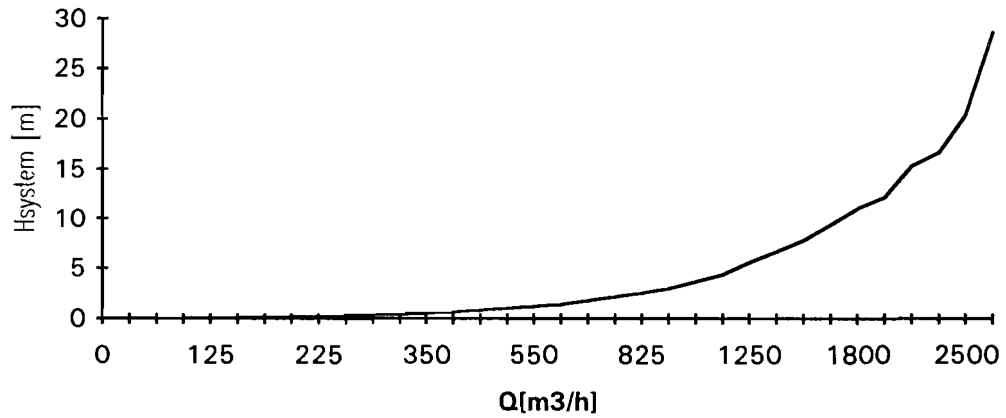


Figure 6.2: $H_{system}(Q_{piers})$ for a tank from Rijk2B.

Of course one wants to check if these calculations are realistic. At the main building of AFS the workstation shows some values of pressures and flows at certain points (only for the situation Rijk to hydrant system). Unfortunately a good measurement can not be made, because:

- There are not enough measuring points. There is not a single one at a hydrant pit. Only in the most valve chambers a pressure measurement takes place (over one valve). The most accurately measurement is 0.1 bar.
- If the system changes quickly (large in-/decrease in flow), it takes a while to establish the new values. So, a good measurement can't be made.
- One does not know when a large flow rate takes place for a longer period of time.

As the duty point of the pump is 127 meter and the $H_{system}(2500)$ is 21,29 meter, one still has to fullfill the necessary pressure at the hydrant. The hydrant needs 104,6 meter and the head produced is 127 - 21,29 = 105,71. From this it can be concluded that the calculations are realistic, because at almost the largest flow the H_{system} equals the $H_{produced}$.

6.6 Piping system tank Oost to tank Rijk

A simpified scheme of the AFS fuel piping system from Oost to Rijk is given in Appendix E2.

6.6.1 Tank 931/933 to pump 901/902

- length 10" pipeline: 50 meter

$$H_v = l \cdot 2{,}15 \cdot 10^{-7} \cdot 10^{(1{,}888\ \cdot \log(Q))} = 1{,}07 \cdot 10^{-5} \cdot 10^{(1{,}888\ \cdot \log(Q))} \quad [m] \qquad (6.6.1.1)$$

- Twin Seal (10x8): $C_v = 3500$

$$\Delta p_{valve} = 1{,}08 \cdot 10^{-6} \cdot (Q)^2 \qquad [m] \qquad (6.6.1.2)$$

6.6.2 Pump 901/902 to the end of the discharge line

- length 8" pipeline: 10 meter

$$H_v = l \cdot 7{,}9 \cdot 10^{-7} \cdot 10^{(1{,}888\ \cdot \log(Q))} = 7{,}9 \cdot 10^{-6} \cdot 10^{(1{,}888\ \cdot \log(Q))} \quad [m] \qquad (6.6.2.1)$$

- TK ball valve (8x6); $C_v = 2022$

$$\Delta p_{valve} = 3{,}22 \cdot 10^{-6} \cdot (Q)^2 \qquad [m] \qquad (6.6.2.2)$$

6.6.3 The end of the discharge line to tank Rijk1B/tank Rijk2B

- length 10" pipeline: 2550 meter

$$H_v = l \cdot 2{,}15 \cdot 10^{-7} \cdot 10^{(1{,}888 \cdot \log(Q))} = 5{,}48 \cdot 10^{-4} \cdot 10^{(1{,}888 \cdot \log(Q))} \quad [m] \qquad (6.6.3.1)$$

- TK ball valve (10x8); $C_v = 4183$
- Twin Seal (10x8); $C_v = 3500$
  $C_{vtot} = 2684$

$$\Delta p_{valve} = 1{,}83 \cdot 10^{-6} \cdot (Q)^2 \qquad [m] \qquad (6.6.3.2)$$

6.6.4 Total losses from Oost to Rijk

The total losses in relation to the flow pumped from Oost to Rijk, obtained from the relations given above, is:

$$H_{tot\ Oost\ Rijk} = H_{vpipeline} + \Delta p_{totvalve} =$$

$$= 5{,}666 \cdot 10^{-4} \cdot 10^{(1{,}888 \cdot \log(Q))} + 6{,}13 \cdot 10^{-6} \cdot (Q)^2 \qquad (6.6.4.1)$$

In table 6.4 the losses for different values of the flow are calculated by means of equation 6.6.4.1. The maximum amount of flow pumped from Oost is equal to 560m^3/h.

Table 6.4: Losses in the system for a given flowrate:

Q [m^3/h]	$H_{vpipeline}$ [m]	$\Delta p_{totvalve}$ [m]	$H_{totOostRijk}$ [m]
50	0,91	0,02	0,93
75	1,97	0,03	2
100	3,38	0,06	3,44
125	5,16	0,1	5,26
150	7,27	0,14	7,41
175	9,73	0,19	9,92
200	12,52	0,25	12,77
225	15,64	0,31	15,95
250	19,08	0,38	19,46
275	22,84	0,46	23,3
300	26,92	0,55	27,47
350	36,01	0,75	36,76
400	46,34	0,98	47,32
450	57,88	1,24	59,12
500	70,62	1,53	72,15
550	84,54	1,85	86,39

Chapter 7

Automated control of pumps

7.1 General

The fundamental control functions can be divided into two groups:
1. the interruption and resumption of flow;
2. the modification of the pump-head capacity-curve.

The first control function can be accomplished by starting and stopping a fixed-speed pump-driver, regulated by means of an on-off control such as a pressure-switch, or an on-off control by a flowmeter. The second control function can be modified by adjusting the speed of a variable-speed pump-driver which is regulated by means of a modulating control such as a constant-pressure control, or a parbolic pressure control.

In paragraph 7.2 the on-off controls will be discussed.

7.2 On-off pump controls

The on-off control system is based on the interruption and resumption of flow. A pump driver is therefore running or not, or a valve is open or closed. This is the simplest closed-loop system to operate on-off between two fixed limits such as water level, pressure or flow.

7.2.1 Pressure switch on-off controls

Depending on technological parameters, the following types of regulation can be distinguished:
1. Simple pressure switch control;
2. Compensated pressure switch control.

Now, these two controls will be shortly discussed and some recommendations will be given.

(a) Simple pressure switch control

This control is based on pump pressure measurements and the main control element is the conventional pressure switch. This regulation offers some advantages and disadvantages:
Advantages:
- simplicity of construction;
- cheap apparatus and relays;
- reduced capital cost.

Disadvantages:
- an extremely high global head range ΔH;
- reduced overall efficiency η_g (40 per cent < η_g < 60 per cent), although the pumps, taken seperately, have high proper efficiencies;
- high specific energy consumption, mainly in the region between the minimum and medium flowrate.

Consequently, the system above is recommended only for the control of pumps with a sloped H(Q) characteristic curve in a pumping installation with a small flow-rate ($Q < 100\ m^3/h$), with the number of pumps $z \leq 2$.

It is obvious that this control is not usefull for the situation Rijkl to hydrant system.

(b) Compensated pressure switch regulation system

The main element of this control is the compensated pressure switch. This control has the following advantages and disadvantages:
Advantages:
- the starting points of pumps with the same characteristic as that of the pumping system become closer, so that starting in ascending cascade is achieved with a reduction of energy consumption in the range of small and medium flow rates;
- a higher overall efficiency of the pumping system (some 10 per cent above that of simple pressure switch regulation) is achieved, since advantageous use of pumps with flat H(Q) characteristics is possible.

Disadvantages:
- it implies additional use of a throttling device;
- the membranes of consumption pressure switches are difficult to pumping systems, having high pumping heads ($H_S \approx 80$-100 meter).

It can be concluded that application of a compensated pressure switch regulation system is recommended for the automated control of pumps with a flat H(Q) curve, fitted to pumping systems with medium flow-rate ($Q \leq 200\ m^3/h$) and small or medium pumping head ($H_S \leq 65$ m).

7.2.2 On-off regulation by flowmeters

Depending on the measured parameter, two on-off regulations can be distinguished:
1. Simple on-off regulation by flowmeters;
2. Regulation by associated pressure switch and flowmeter.

(a) Simple on-off regulation by flowmeter

In this regulation the pumps are used in accordance with the flow-rate. It requires at least two pumps in the pumping installation. Depending on their entrance sequence into the operation cycle, one pump is called *the lead*, while the other(s) is(are) called *the lag* pump(s). The lag pumps are identical, but the lead pump may have a flow-rate, equal to or half that of the others. When the lead pump has a smaller flow-rate, it is also called a *jockey* pump, while the others are called main pumps.

The on-off regulation by flowmeters is based on meeting all consumption needs by utilising only the pump operation ranges. Consequently, for each requested flow-rate in the system, there is a corresponding stable characteristic point on the H(Q) curves of one or more pumps, operating in parallel. When the requested flow-rate is increasing, the operation point describes the characteristic curve of the pump or pumps. When the flow-rate reaches the value at which the discharge pressure equals the system characteristic pressure, the flowmeter always gives a signal for the starting of an additional pump. Consequently, the pressure rises rapidly to a new value, corresponding to the new flow-rate, requested in the system. A flow-rate decrease determines a reverse displacement of the duty point, and then the flowmeter controls the pump stopping. Figure 7.1 shows the operation of such pumping systems.

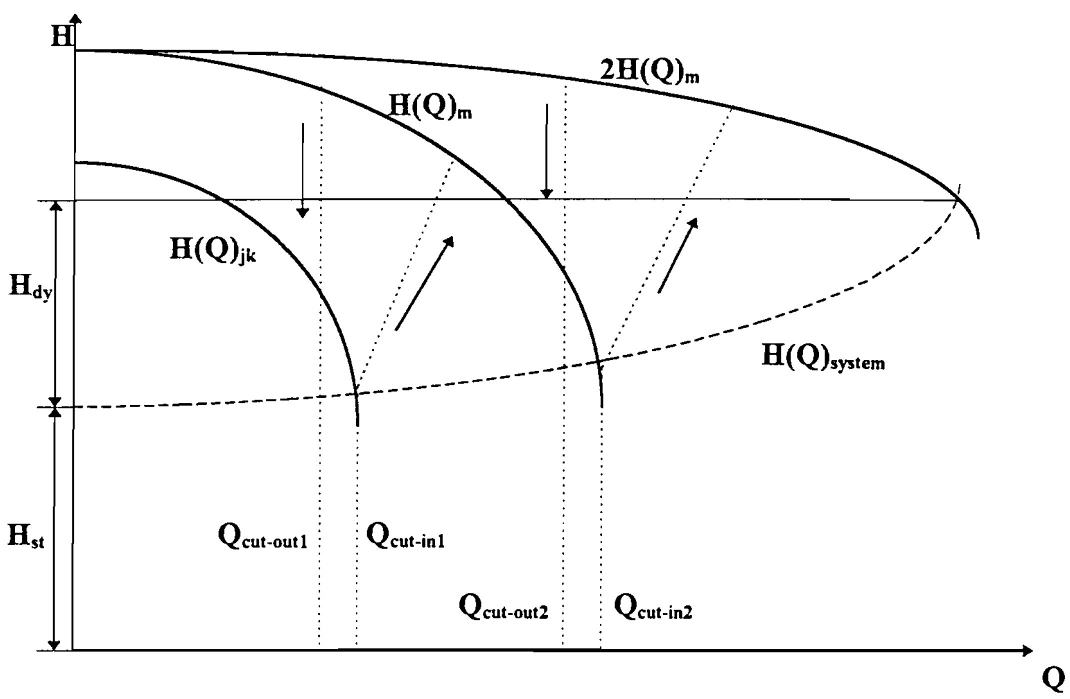


Figure 7.1: Head-discharge curve of two duty pumps regulated by flowmeter.

The on-off regulation has the following advantages and disadvantages:
Advantages:

- the great advantage of this system is the continuous operation of the pumps at all flow-rates, higher than the cut-out flow-rate of the first pump, which reduces equipment fatigue and extends the life span of machines (pumps and electric motors);
- the system achieves a high efficiency of the pumping installation due to the use of pumps with a flat H(Q) characteristic.

Disadvantages:

- the system implies the use of flowmeters whose throttling device leads to high head losses;
- it requires the use of a stand-by pump with automated control by pressure switch, which should take place of any duty pump in trouble and thus avoid the breaking of the flowmeter chain.

Consequently, the system analysed is suitable for automated control of pumps with a flat H(Q) characteristic curve, mounted on medium flow rate pumping installations ($Q < 720\ m^3/h$).

(b) On-off regulation by associated pressure switch and flowmeter

This system, like pressure switch on-off regulation, gives an intermittent operation of pumps, the consumption range being covered by pumps with hydropneumatic tanks. These tanks are brought into use when the requested flow-rate is beyond the individual operation ranges of the pumps with which they work. The pressure head in the tanks varies between two limits, H_{cut-in} and $H_{cut-out}$, which are rather close to each other.

This regulation has the following advantages and disadvantages:
Advantages:
- it is very simple constructed and has a simplified relay;
- a high overall efficiency;
- the flowmeter used must not necessarily be of high accuracy, since only correct estimates of the flow-rate range is interesting.

Disadvantages:
- In certain cases, namely when the number of pumps z is too large ($z > 5$), the system implies additional relays.

Thus, it can be concluded that this system is all the more adequate as it has a flatter characteristic curve. Therefore, it was developed considerably for the case of automated control of pumping installations for putting under pressure spraying irrigation system, whose $H_S(Q)$ characteristic is a horizontal line, parallel to the flow-rate axis. Consequently, the cut-in pressure of all pumps is settled equal to the system-head curve.

7.3 Modulating pump controls

7.3.1 General

The modulating control system is based on modification of the pump head curve by means of speed variation. It adjusts the speed of pump driver to current needs of the pumping system. This is the best type of closed-loop control.

The commands given by these controls are continuous, since their action has a character, leading to continuous changes in the operation of some machines (i.e., motor speed variation), which continuously change the installation parameters (flow-rate, pressure, etc.).

In modulating regulation systems, the final control element is the variable-speed pump. These systems are used for automatically creating a certain dependence law between pump pressure and flow-rate so that their H(Q) pump-head curves should be ideally placed on the path followed by the $H_S(Q)$ system-head curve. The $H_S(Q)$ is represented by a parabola with its concavity upwards (in the case of water supply systems), or by a horizontal line, parallel to the flow-rate (in the case of land irrigation systems). Unlike these curves, the natural characteristic curve of a pump with constant speed is represented by a parabola with downward concavity, which is opposite to that of the system.

This difference is a great disadvantage in the case of pump flow-rate regulation from the nominal point up to the region of minimum flow-rates, since in this region the pump pressure far exceeds the necessary pressure demanded by the system-head curve. This leads to useless hydraulic energy losses, and thus, to a rather low global efficiency of pumping installation. For preventing this shortcoming, one should use variable speed pumps.

By changing the speed of these pumps, a family of pump-head artificial curves, H(Q), is obtained. The hull of these curves could be (depending on imposed speed law) either a parabola with upward concavity, or a horizontal line, which should superpose exactly over the $H_S(Q)$. Thus, ideally speaking, one may obtain:

- the head, developed by the pump, is equal to the system head characteristic curve;
- the global efficiency of the pumping installation is excellent.

Now, the analysis of modulating regulating systems as applied to pump controls will be discussed.

7.3.2 Pressure modulating regulations

These regulations have the role of creating a dependence law between pump head and flow-rate, when the demand in the system is changed. Essentially, two types of regulations are used:

1. constant pressure regulations;
2. parabolic pressure regulations.

7.3.2.1 constant pressure regulation

This regulation measures a single hydraulic parameter: the outlet pressure of the pumping station. The role of this regulation is to keep constant the pump outlet pressure, regardless of variations in consumption or/and inlet pressure.

In figure 7.2 a schematic diagram of a constant pressure system is given.

In this figure the following abbreviations are used:

VSM: the variable-speed motor;
VSP: the variable-speed pump;
Tp: continuous pressure transducer;
Cp: pressure controller;
TS: temperature sensor;
SV: solenoid valve;
FC: frequency converter.

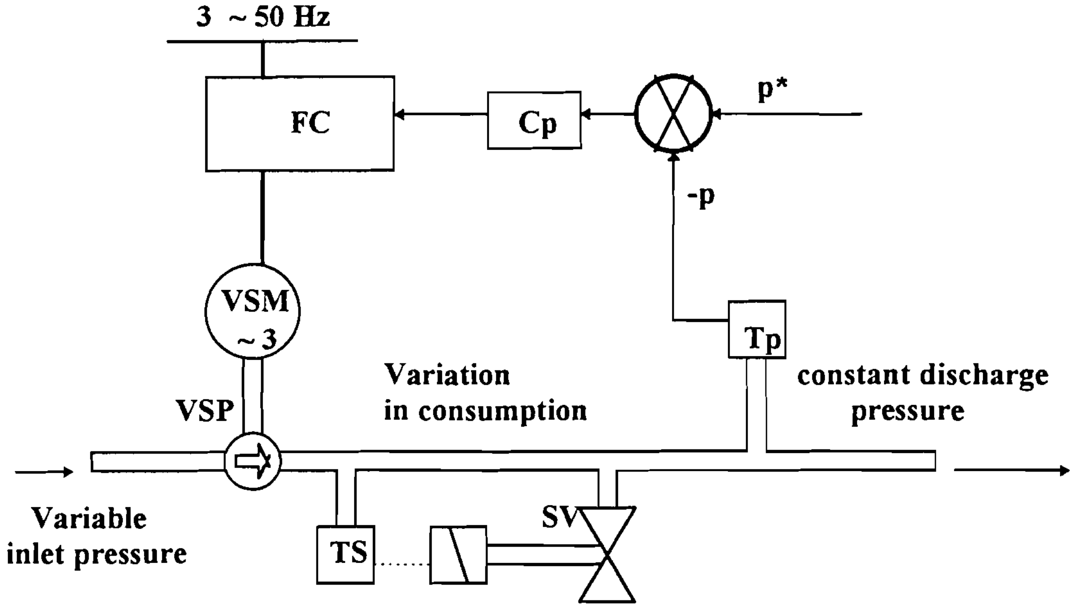


Figure 7.2: Schematic diagram of constant pressure regulation with variable-speed pump.

This system uses Tp as an element of transmitting information which converts the hydraulic pressure signal into an electronic one, p, and, at the same time, informs Cp, on the real value of pressure. Cp compares the real value, p, with the setting value, p*, and gives the order to actuation element, FC. This order is transmitted to the VSM as a speed variation, so that the VSP should adequately change its flow-rate, thus keeping constant discharge pressure.

Now, the following cases will be discussed:

- with only one variable-speed pump;
- with two pumps, of which the first with variable speed and the second with fixed speed.

(a) *one variable-speed pump*

Here, the pump operation depends on the disturbances occurring in discharge pressure. It takes place as follows:

> when pressure falls under the preset value, p*, due to demand increase in pumping system, the controller makes the driving motor rise its speed, and the pump, its flow-rate, so that the discharge pressure could be kept constant. Conversely, when discharge pressure rises, tending to exceed p*, due to demand falling down, the controller orders a speed lowering, and implicity, a lowering of pumped flow-rate.

Figure 7.3 shows the operation diagram of a pump with constant pressure regulation. The artificial characteristic curve $H(Q)_{VSP}$, obtained by regulation, is represented by the *AB* segment and has the form of a straight line with constant ordinate $H_{ds} = H_{max}$ = constant (parallel to the flow-rate axis). The speed variation range, Δn, is narrow, and the control range of flow-rate, ΔQ, is rather limited.

The limitation of ΔQ range is determined by the fact that in minimum flow-rate region the overall efficiency of the installation is rather low. For widening the control range of ΔQ_i, the peak demand of pumping system should be distributed to two or more pumps, connected in parallel, of which one with variable speed.

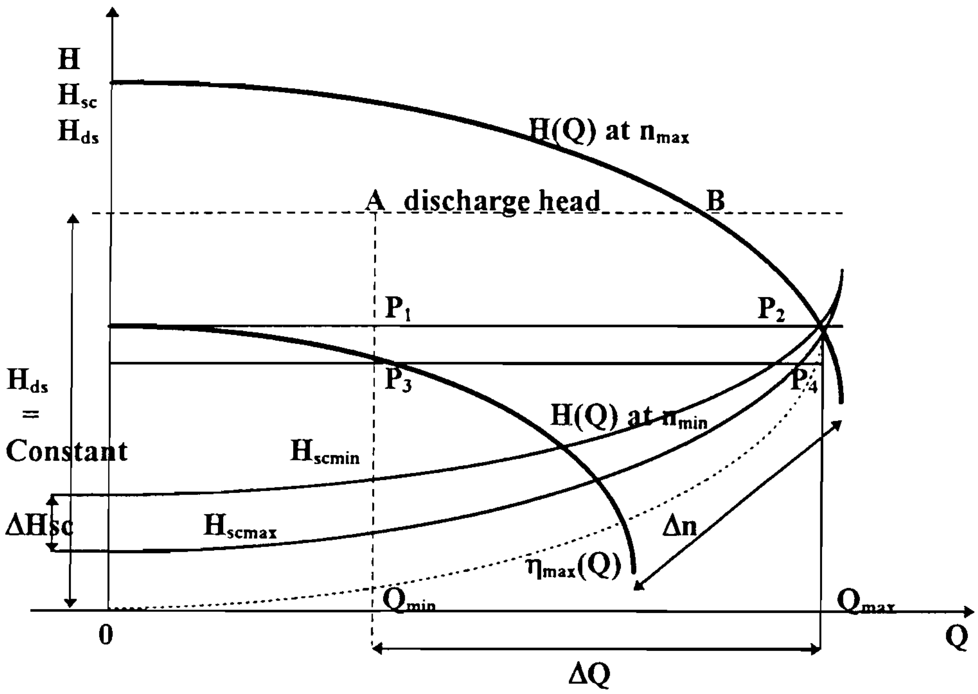


Figure 7.3: Head-discharge curves of a variable speed pump, regulated by constant pressure system. The square P_1, P_2, P_3 and P_4 is the working range of the pump.

(b) *two pumps, of which one variable and the other fixed*

In this case, the resulting artificial characteristic of parallel clutch is obtained by the continuous speed modification of variable speed pump, VSP, and by starting and stopping the pump or pumps with fixed speed, FSP.

Figure 7.4 shows the operation of the VSP and FSP, connected in parallel and regulated by constant pressure system. The artificial characteristic obtained (the AB segment) is also represented by a horizontal line, parallel to the flow-rate axis. These pumps operation depend also on the disturbances, occured in discharge pressure as a result of demand modification, and takes place as follows:
The first pump, VSP, covers by itself the left part of demand variation range, ΔQ, keeping constant its pressure by changing pumpled flow-rate through speed variation.

When the demand exceeds the pumped flow-rate, secured by this pump at its maximum speed, then the FSP is started by means of a maximum voltage relay, connected to the electric circuit of a voltage tahogenerator. When this latter pump starts running, the VSP is immediately and adequately lowering its speed, meeting the new demand. For covering the right side of the ΔQ range, the FSP settles a certain duty point (for instance, corresponding to maximum efficiency), while the VSP duty point is displaced (for coping up with consumption variations) by the modification of pumped flow-rate and speed, but keeps its pressure constant. When the demand falls down under a smaller value than that of the flow-rate, pumped by the two pumps (with VSP working with minimum speed), the FSP is stopped by means of a minimum voltage relay, and the VSP rising immediately rising its speed for meeting its new demand.

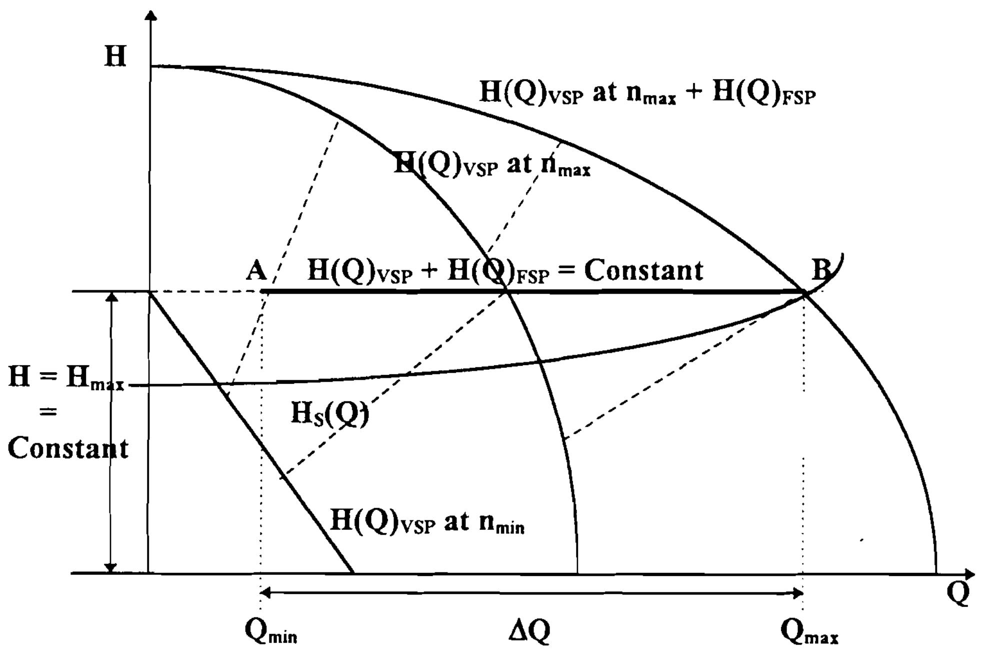


Figure 7.4: Head-discharge curves of two pumps regulated by constant pressure system.

The regulation presented above has the following advantages and disadvantages:
Advantages:

- it keeps a constant outlet pressure, regardless of variation in consumption or/and inlet pressure.

Disadvantages:

- for a small number of pumps, it covers a reduced flow-rate range, since the characteristic resultant due to pump regulation has a path (in region of minimum demand) far away from that of maximum efficiency characteristic curve belonging to variable-speed pump.

Consequenly, the constant pressure regulation is recommended for the automatization of pumps operating on horizontal system-head curve $H_S(Q)$.

7.3.2.2 Parabolic pressure regulation

This regulation is simultaneously measuring two hydraulic parameters:
1. the discharge pressure;
2. the flow rate (equal with the consumption in system).

The role of this regulation is to automatically establish a parabolic dependence between pump head and flow-rate, to obtain an artificial pump-head curve, H(Q), under the form of a parabola with upward concavity, superposing over the system-head curve, $H_S(Q)$, with natural parabolic paths.

Figure 7.5 shows the simplified diagram of a parabolic pressure system. This system combines a pressure transducer, Tp, with a flow-rate transducer, Tq, both connected to an adder instrument. The hydraulic signals of pressure and flow-rate are thus converted into a resulting electronic signal of compensated pressure, by flow rate p_c.

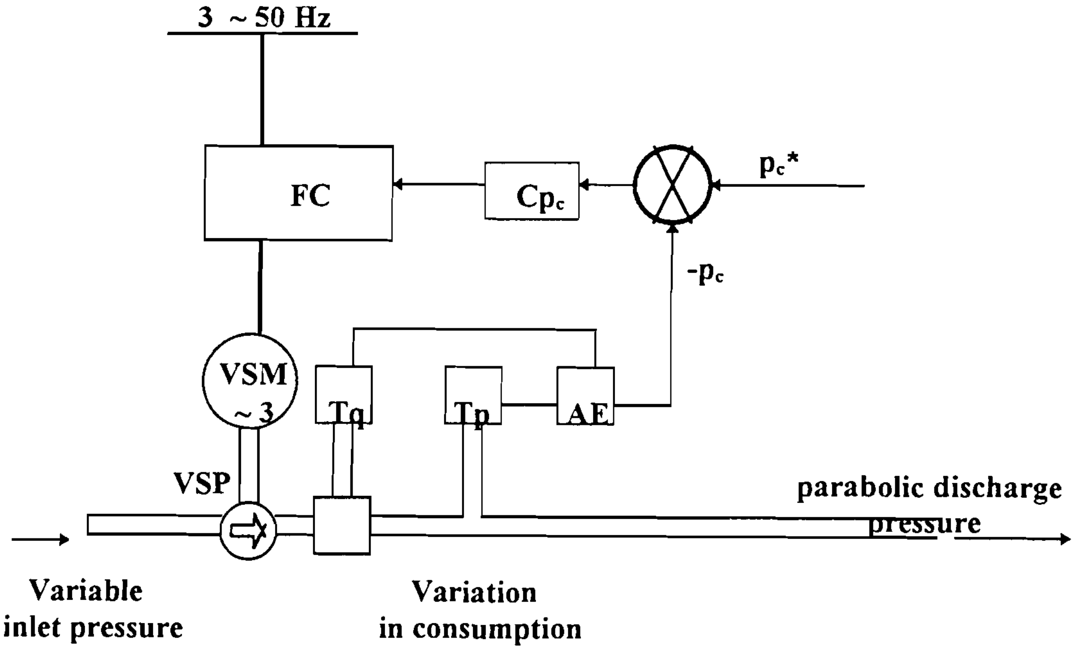


Figure 7.5: Simplified diagram of a parabolic pressure system.

In figure 7.5 the following abbreviations are used:
VSM: the variable-speed motor;
VSP: the variable-speed pump;
Tp: pressure transducer;
Tq: flow transducer;
AE: added element;
Cp_c: compensated pressure controller;
FC: frequency converter.

Thus, the transducers inform Cp, on compensated pressure real value. This time, the controller compares real value p_c, with preset value p_c*, and gives the adequate order to the actuating element FC. This command is transmitted to the VSM as a speed variation, so that the VSP should achieve a reversed parabolic dependence between pressure and flow-rate, similar to that of the pumping system.

The number of pumps depends on system static head. Thus, when the pumping system is lacking static head, then the artificial pump-head characteristic curve, $H(Q)_{VSP}$, is totally superposed over the maximum efficiency curve, $\eta_{max}(Q)$, and therefore one single pump is enough for seccuring an excellent overall efficiency to pumping installation. When static head is present, and the system-head curve starts flattening, the peak demand should be secured by two or more pumps, connected in parallel, of which a minimum of one should have variable speed.

The pump-head characteristic of the two pumps has been drawn in figure 7.6 and is represented as the *AB* curve segment. It is noticed that the resulting characteristic curve $H(Q)_{VSP}$ + $H(Q)_{FSP}$ also has a parabolic shape, superposing exactly over the system-head curve $H_S(Q)$. The connected pumps operation is similar to that described in case of constant (horizontal) pressure regulation.

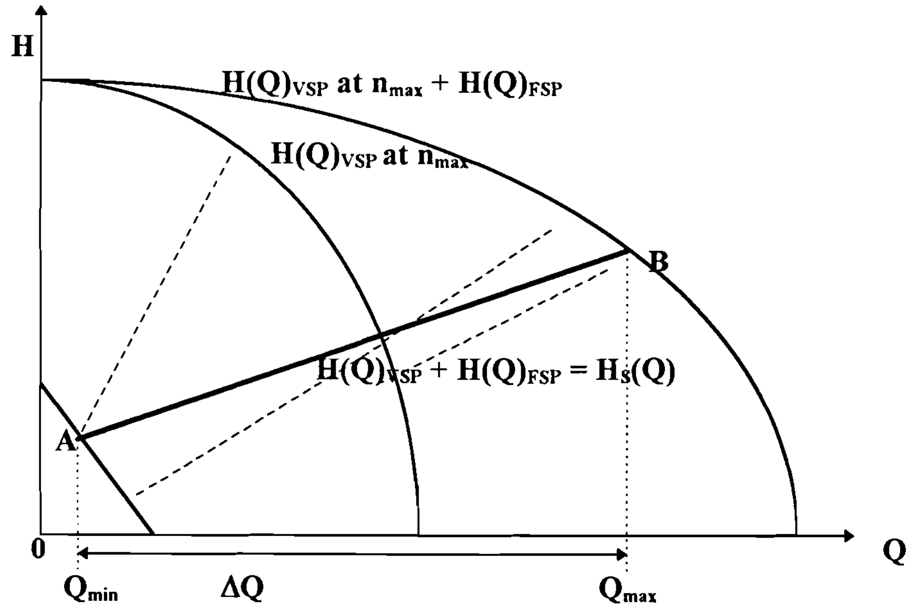


Figure 7.6: Head-discharge curves of two pumps, regulated by parabolic pressure system.

The parabolic pressure regulation has the following advantages and disadvantages:
Advantages:
- the artificial pump-head characteristic curve is exactly superposed over the system-head curve, thus securing an excellent efficiency of pumping installation;
- the range of covered flow-rate is very large, since the artificial pump-head characteristic has a common or close path to maximum efficiency characteristic.

Disadvantages:
- it makes use of a more complex regulation system, since it requires two transducers and an adder instrument.

Consequently, the parabolic pressure regulation is recommended for the pump operation on inclined system-head characteristics. These are the cases of distribution pumps, connected to long main lines with high dynamic head.

7.4 Situation Rijk to hydrant system

In figure 7.7 the situation at AFS for the pumping from Rijk to the hydrant system is given.

- The average height of the tank is 8 meter (see also paragraph 12.5).
- A new pump is started when the flow exceeds (the amount of pumps already working) times ($Q_{pompmax}$ = 288 m^3/h).
- At the hydrant pit a pressure of 8,5 bar (104,6 meter) should be present.

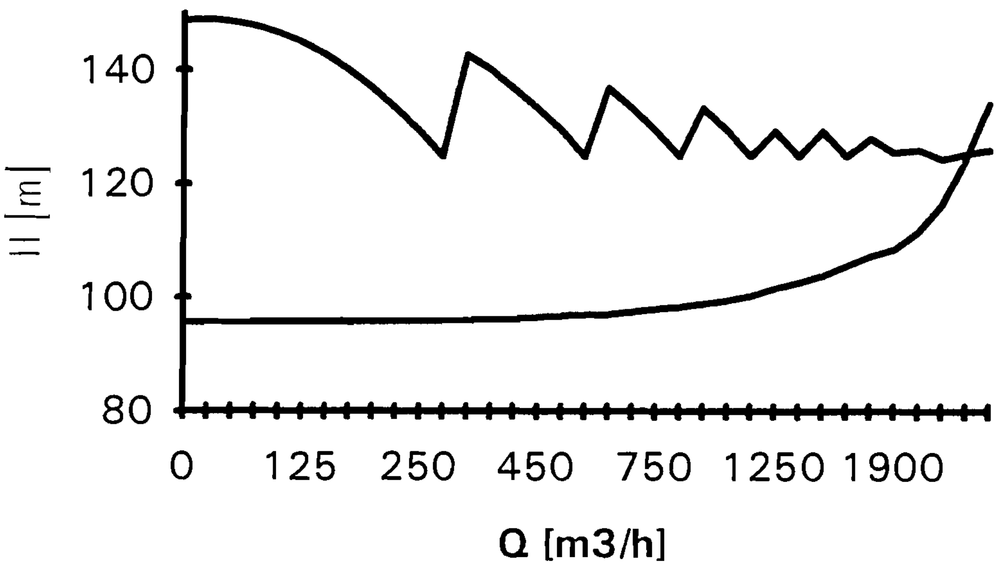


Figure 7.7: H_{system} (static:8,5 bar and dynamic:losses(Q)) and $H_{produced}$ by the pumps for the pumping Rijk to hydrant system.

From figure 7.7 it is noticed that especially at 'small' flows too much head is produced, so energy is wasted. By using speed control this difference can be reduced and energy saved. First the speed control will be dealt with. After that the energy savings will be calculated.

Chapter 8
Variable Speed Drives

8.1 Introduction

In the past, the traditional technique of controlling electric Variable Speed Drives (VSD's) relied on DC motors equipped with carbon brushes, which proved to be very unreliable and maintenance intensive in a refinery environment. As a consequence, the incentive for their installation diminished until the late 1970's, at which point powerful semiconductors became available which could handle larger currents at higher voltages. Power converters with controllable output frequency and voltage were therefore available to drive a squirrel cage AC motor which had already proven to be the most reliable motor with little maintenance required. With soaring energy prices in the early 1980's, VSD's proved to be very successful in saving energy and controlling the process with respect to conventional throttle control [Dronoff et al., 1990].

One of the previous constraints of electric VSD selection was the maximum rated power and speed possibilities. Present VSD technology shows that these constraints are no longer applicable.

The bottleneck in the selection of electric VSD's over fixed speed is still the significantly higher price, which was in 1990 for installed equipment about 2,0 times the price of a similar sized fixed speed drive [Dronoff et al., 1990]. If one wants to install VSD's, it certainly has to be justified. That's why each application has to be evaluated with respect to the operating envelope required as well as the overall energy balance of the particular operating company.

One can distinguish the following options:

- fixed speed with throttle (reduce impeller size at lower throughputs if limited rangeability is not constraining);
- inlet vane control (poor efficiency at low loads);
- Speed control can be achieved by means of the following types of mechanisms:
    1. Transmission mechanisms:
        - Eddy current coupling
        - Fluid coupling;
    2. Adjustable voltage a.c. drives;
    3. Wound rotor induction motors;
    4. D.C. motors;
    5. Variable frequency drives

These different speed control mechanisms will now be discussed.

8.2 Variable-speed transmission mechanisms

By these mechanisms a transmission element is inserted into the drive train. The eddy current coupling and the fluid coupling, used for this purpose, are made of two members that are mechanically independent:

1. the input member, or the primary, connected to the motor;
2. the output member, or the secondary, connected to the load (pump).

By this arrangement the speed of the primary is kept constant and that of the secondary can be progressively and continuously varied.

8.2.1 The eddy-current coupling

This coupling is an electromechanical torque-transmitting device, which transforms torque T_1, speed n_1, and power P_1 from the primary into T_2, n_2, and P_2 in the secondary, of different magnitudes from those of the primary. A typical self-contained air-cooled eddy-current coupling is shown in figure 8.1 [Ionel, 1986].

The input and output members are mechanically independent, with the output magnet member revolving freely with the input ring or drum member. An air gap separates the two members, and a pair of antifriction bearings serve to maintain their proper relative position. The magnet member, placed in the exterior, is called the armature circuit. It is made from a solid iron cilinder *1*, and of a driving shaft *5*, mechanically connected to the drive shaft *6*, which rotates it at constant speed. The input ring, placed inside, is called the inductor and is made of the rotor *2* (laminated with grooves for the excitation winding *3*) and the led shaft (secondary) *6*, on which the slip rings *4* are fixed for electrical connection to the source of direct current.

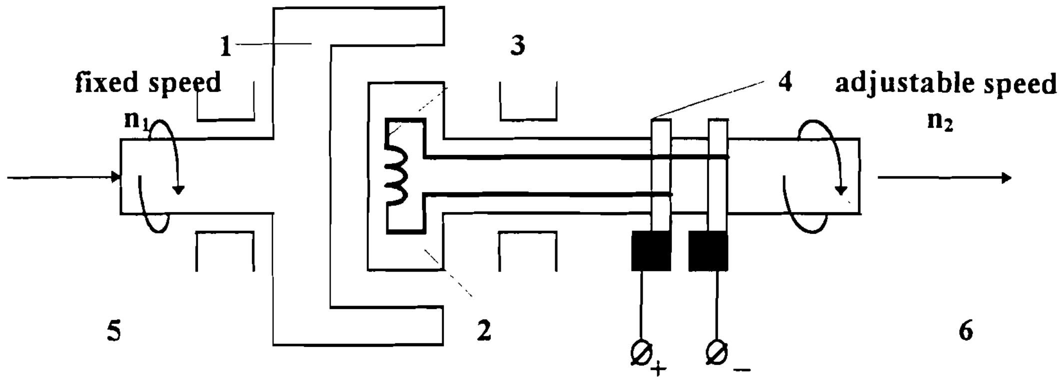


Figure 8.1: Schematic diagram of an eddy-current slip coupling.

Generally, two quantities should be determined for computation of an eddy-current coupling:
1. the slip power;
2. the coupling efficiency.

(a) *The slip power*

Since the input power $P_{ECC\,i}$, received from the primary shaft of the electromagnetic coupling, is proportional to torque T_1, developed by the electric motor, and to that of the speed n_1 of the motor, it follows from equation 8.2.1.1:

$$P_{ECCi} = \frac{T_1 \cdot n_1}{9545} \tag{8.2.1.1}$$

The power output, transmitted to the secondary shaft, $P_{ECC\,o}$ is proportional to the driving torque T_1, transmitted by the electromagnetic coupling, and to the speed of the secondary shaft n_2. Thus

$$P_{ECCo} = \frac{T_1 \cdot n_2}{9545} \tag{8.2.1.2}$$

The slip power of the eddy-current coupling $P_{ECC\,s}$, which is lost in the coupling, is defined as the difference between the power input and output:

$$P_{ECCS} = P_{ECCi} - P_{ECCo} = \frac{T_1 \cdot n_1}{9545} \cdot \left(\frac{n_1 - n_2}{n_1}\right) = \frac{T_1 \cdot n_1}{9545} \cdot S_{ECC} \tag{8.2.1.3}$$

where S_{ECC} denotes the coupling slip.

(b) *Coupling efficiency*

By not considering the losses, the efficiency is expressed by

$$\eta_{ECC} = \frac{n_2}{n_1} \tag{8.2.1.4}$$

When the losses (frictional losses occurring in bearings, ventilation losses, and the losses for excitation) are considered, which together represent some 2 per cent of the nominal power and are proportional to the speed, equation 8.2.1.4 becomes

$$\eta_{ECC} = \frac{n_2}{n_1} - 0{,}02 \cdot \frac{n_2}{n_1} = 0{,}98 \cdot \frac{n_2}{n_1} \tag{8.2.1.5}$$

The total efficiency of a driving installation, provided with an eddy-current coupling, $\eta_{tot\,ECC}$, is given by the equation

$$\eta_{tot\,ECC} = \eta_m \cdot \eta_{ECC} \tag{8.2.1.6}$$

It is noticed that at small speeds n_2 and large slips s, the efficiency $\eta_{tot\,ECC}$ of a driving installation, provided with eddy-current coupling, is rather low and is decreasing approximately linear with speed. Thus, speed control of pumps by means of eddy-current couplings is not economical in energy terms.

Driving installations, fitted with eddy-current couplings, have the following advantages and disadvantages [Ionel, 1986] and [Rayner, 1995]:
Advantages:
- they have a simple construction, and hence their cost is reduced;
- they can drive pumps with high moments of inertia (developed at starting) without overheating the starting cage;
- they require low commands powers, allowing at the same time easy coupling and decoupling of shafts;
- they secure the possibility of using an asynchronous motor for high power pumps.

Disadvantages:
- they have a low efficiency at small rotation speeds, due to the rise of slip losses;
- they require complex exploitation, because the eddy-current coupling is an electrical machine with two rotating parts;
- their speed-torque characteristics show low rigidity and thus reduced stability.

8.2.2 Fluid coupling

The fluid coupling consists, put simply, of a pump impeller on the driving shaft and a turbine runner on the output shaft. The input torque is transmitted to the output shaft by the mass forces of a fluid (mostly oil) flowing between pump impeller and turbine runner (see figure 8.2) [Sulzer, 1994].

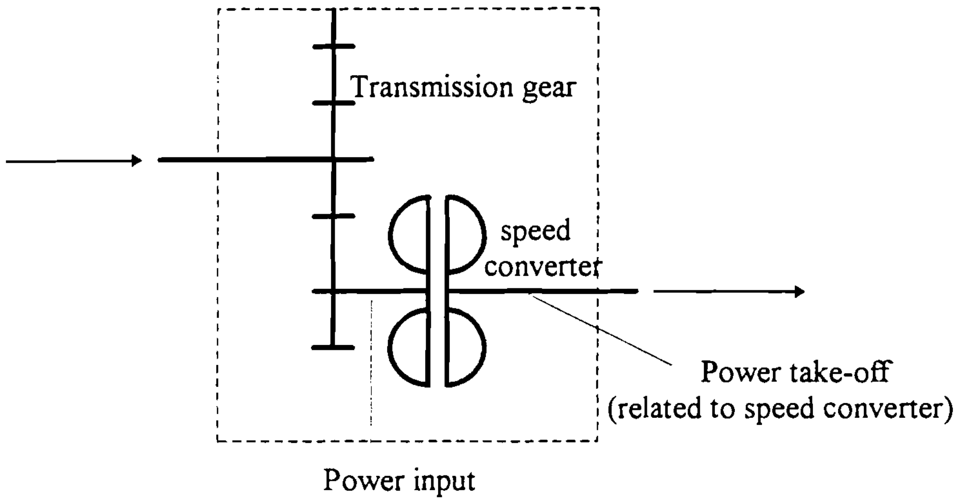


Figure 8.2: Schematic diagram of a fluid coupling.

The slip is equal to

$$s = \frac{n_1 - n_2}{n_1} = 1 - \frac{n_2}{n_1} \tag{8.2.2.1}$$

If the very small torque loss due to air and bearing friction is neglected, the input and output torques are equal. The following relation is obtained for the torque:

$$T_1 = T_2 = T = K \cdot \rho \cdot n_1^2 \cdot D_p^5 \tag{8.2.2.2}$$

where K = Characteristic comprising the constant factor related to the chosen dimensions and the function of all the other parameters;
ρ = Density of operating fluid;
D_p = Converter size.

Due to hydraulic and power losses, the power at the secondary shaft (P_2) is always smaller than that at the primary shaft (P_1). This is the reason for the occurrence of the slip effect during torque transmission between runner and pump impeller. For the P_1 and P_2 the following relations are given

$$\begin{aligned} P_1 &= T \cdot \omega_1 \\ P_2 &= P_1 \cdot \frac{n_2}{n_1} = P_1 \cdot (1 - s) \end{aligned} \tag{8.2.2.3}$$

The transmission efficiency is given by

$$\eta = \frac{P_2}{P_1} = \frac{n_2}{n_1} = 1 - s \tag{8.2.2.4}$$

The power loss due to heatup is proportional to the slip:

$$P_v = P_1 - P_2 = P_1 \cdot s = P_1 \cdot \frac{n_1 - n_2}{n_1} = P_2 \cdot \frac{n_1 - n_2}{n_2} \tag{8.2.2.5}$$

If all losses are taken into account, the P_1 and P_2 are related as

$$P_1 = \frac{P_2 \cdot \left(1 + a \cdot \frac{n_1 - n_2}{n_2}\right) + b}{y} \tag{8.2.2.6}$$

and the power loss is equal to

$$P_v = \frac{P \cdot n^2}{y} \cdot |a - A \cdot n| + B \tag{8.2.2.7}$$

where A = a + y − 1;
B = b / y;
a = 1.12 - 1.18 = coefficient of power lost in the oil circuit and mechanical losses;
b = 10 - 50 kW = power required for auxiliary oil pumps (operating and lubricating oil circuits);
y = 0.97 -0.985 = efficiency of step-up gear (if fitted);
a, *b*, and *y* are given by the supplier;
P = rated pump output converted to the speed n_1 (with slip = 0)

$$P = P_2 \cdot \left(\frac{n_1}{n_2}\right)^3$$

$$n = \frac{n_2}{n_1}$$

This means that for a specific speed converter the lost power P_v becomes a maximum when values are as follows:

$$\frac{n^2}{y} \cdot (a - A \cdot n) = Maximum$$

This is fulfilled when:

$$n = \frac{2 \cdot a}{3 \cdot A}$$

and therefore:

$$P_{v\max} = P \cdot \frac{4 \cdot a^3}{27 \cdot A^2 \cdot y} + B \tag{8.2.2.8}$$

The above equations correspond to the operation of a pump with origin parabola as resistance curve (pipe resistance characteristics). When the resistance curve is flat (static and dynamic head) the load variation must be taken into account.

The fluid coupling has the following advantages and disadvantages [Sulzer, 1994]:
Advantages:
- Stepless control in a range of 4:1 to 5:1 maximum;
- Capable of transmitting very high power outputs;
- Soft start acceleration, Gentle pump acceleration;
- Load free motor starting;
- Simple torque matching by modifying the oil fill;
- Assured protection against excessive heating by means of fusible cutout;
- High efficiency as rated slip is very low.

Disadvantages:
- Price;
- Additional space requirement for the converter and its auxliaries, such as cooler etc.;
- Rated capacity of electric motor slightly increased to compensate for converter losses (rated slip and mechanical los generators such as bearing and gear train driving power for oil pumps);
- Low efficiency when the slip is not low. This is akin to throttling.

8.3 Adjustable voltage a.c. drives

Adjustable voltage a.c. drives consist of a high-slip motor (10%), constant frequency adjustable voltage control system.They are available in capacities up to 60 kW [Rayner, 1995].

Considering drive oversizing and to the losses inherent in the control feature, speed control based on change in the supply voltage is only economical, when it is applied to low power pump electric drives ($P \leq 5$ kW).

Another, more economical solution, that does not call for an oversize motor, consists in employing a disk-rotor (axial-gap) asynchronous motor of a special design. The resistance of the disk-rotor varies in inverse proportion to the speed. The disk-rotor asynchronous motor, supplied through a thyristorized variator, represents an economic solution for the adjustable drives of low power pumps ($P \leq 10$ kW), operating in adverse environmental conditions (such as chemical and petrochemical plants) [Ionel, 1986].

8.4 Wound rotor induction motors

Wound rotor induction motors together with a rheostat or other solid state secondary resistance control provide a means for soft start variable speed control that has found frequent use with pumps, especially in waste water applications.

They have been available up to 2250 kW in all standard voltages. Except in the small sizes, this drive has been the most popular in the past with

- good functionality;
- a relatively low first cost.

8.5 DC motors

The finest speed control is obtained by means of d.c. motors, fed from variable voltage supplies. Voltage controlled electric drives confer an economic speed control over a wide enough speed range (Δn = 1:200) and yet are used with pump drives only in special cases, because

- they are expensive due to
  - the fact that the d.c. motor is more expensive than the a.c. one;
  - the static converter, through which the motors are supplied.
- they are difficult to operate (refers to d.c. commutator motors).

The control procedure, changing the voltage applied across the armature while keeping the excitation current at a constant value, is preferred with pumps with a parabolic mechanical characteristic (i.e. their resistive torque varies in direct proportion to the square of the speed) and consequently they require the highest torque at the highest speed.

The converter, used to supply a variable voltage to d.c. motors, are converters with external (natural) commutation. Their basic function is to convert the electric energy, of a variety of kinds and parameters, by means of natural commutation. This external commutation can be achieved either from the main supply or from the load (the motor), and hence converters are classified in two groups:

1. phase-controlled converters;
2. load-controlled converters.

The adjustable drives of d.c. motors were developed along these general lines [Ionel, 1986].

8.6 Variable frequency drives

The angular velocity of the asynchronous motor is:

$$\Omega_2 = \Omega_1 \cdot (1-s) = \left(\frac{\omega_1}{p}\right) \cdot (1-s) = \left(\frac{2 \cdot \pi \cdot f_1}{p}\right) \cdot (1-s) \tag{8.6.1}$$

From this relation, it readily follows that one possible way of speed control is by changing the frequency f_1 of the voltage, applied across the stator windings. This method applies a frequency converter, since the industrial mains are fed at a constant frequency, and therefore requires significant investment costs, for a converter is generally more expensive than is the asynchronous motor whose speed is to be adjustable.

Variable frequency drives (VFD's) have been in use for over ten years. They provide soft starts and very good efficiencies at reduced speeds. Of late, advancements in the technology of these drives and substantial reduction in manufacturing cost have made them the preferred choise. Standard pre-engineered units are now available up to the 450 kW sizes, with custom engineered sizes available up to three times that high. Payback periods are now in the eight month to two year range [Baljevic, 1993].

Keeping a constant voltage to frequency ratio when changing frequency maintains a constant flux in the motor and full torque capability.

$$V / Hz = \text{constant}$$

Holding this ratio, results in the motors power capability changing directly with speed (3.7.1.1).

Under these conditions, the current will remain constant as the speed is changed.

The type of load is an important factor in selecting the VFD. Variable torque VFD's, as pumps impose variable torque loads ($T \propto N2$), will have efficiencies in the 96-97% range at full load and drop down to only 86-87% at 15% load, which is a very substantial improvement over throttling and slip drives. They can be overloaded by 110% for one minute. In figure 8.3 the head savings with speed control by VFD following the system curve vs a trottling valve following the HQ characteristic curve for any given capacity is given.

Multiplying this by the VFD efficiency and the time spent at each capacity over a period of time such as a say will give the power savings over that period.

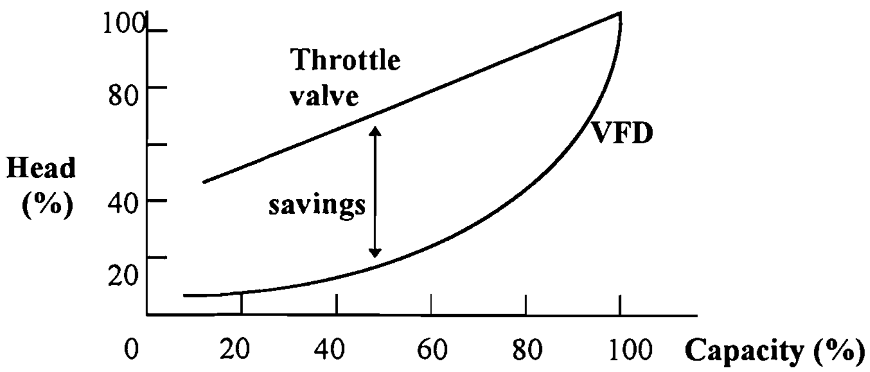


Figure 8.3: VFD head savings versus capacity. Reprinted from [Rayner, 1995]

Both hydraulic and eddy-current couplings can be utilized to achieve adjustable-speed operation. The modified pump curves are the same as for ac adjustable-speed drives. The advantage of VFD's is that they remain fairly efficient across a wide speed range whereas an adjustable-speed coupling has a value of efficiency that decreases directly with speed. A typical efficiency curve for an VFD and an eddy-current coupling is given in figure 8.4.

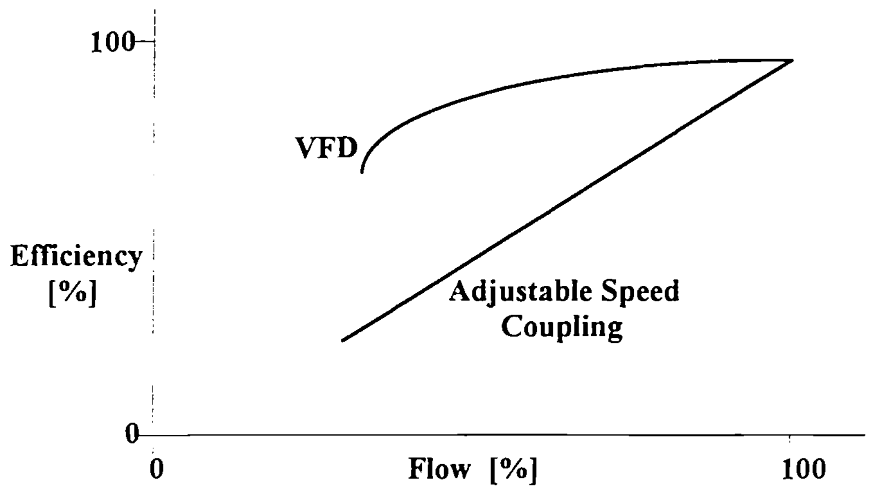


Figure 8.4: Typical efficiencies for several means of flow control.

If the speed turndown requirement is small, the coupling slip losses may not be excessive. However, some users have found adjustable-speed couplings to require excessive maintenance, and have made this the chief justification for the use of VFD's in addition to the energy savings [Rice, 1988].

The following recomendations are from VFD manufactures:
- Inverter duty motors are now available for VFD's. Conventional motors can be used but they must be selected with some care. High Efficiency Rated motors are preferable;
- For pumps with their variable torque loads use a motor with a 1.15 service factor, but do not use the service factor;
- The design B motor is best suited for VFD's;
- Startup torque requirements must be watched closely, i.e. positive displacement pumps are likely candidates for a larger size VFD;
- There are elevation and temperature deration factors for VFD selection;
- Isolation transformers may be necessary due to the harmonics generated;
- Vertical pump/motor assemblies must be checked for reed critical natural frequencies when applied to variable speed.

VFD/pump drives have the following advantages:
- Significant power savings;
- Soft start;
- Starting power penalties eliminated or reduced;
- Possible rebates from utilities for power savings;
- No starter or throttle valve required;
- Increased pump life due to reduced recirculation and soft starts;
- On multiple pump control, the need for pump over-rated capacity when fewer pumps are on can be eliminated. In some cases one less pump may be required.

8.7 Conclusions

Frequency controlled electric VSD, so a VFD is the *best* solution with respect to rangeability, speed of response, control accuracy and maintenance. They offer the advantage of soft restart of the electric motor after a power failure and contrary to fixed speed motors no high current is drawn from the electric network during start/restart. Another advantage is the easy reversal of the motor's direction of rotation and that the speed reference signal can include both directions of rotation.

One can say a lot about the pro's and con's of VSD's with respect to fixed speed, but the best final solution, however, will be achieved by a combined approach of operations, technology, process control and engineering to ensure that the real process objective is met.

In the next chapter the VFD will be discussed in detail.

Chapter 9
Variable Frequency Drives

9.1 General

Proper application of controllers to standard induction motors to obtain variable-speed operation requires a detailed understanding of the application requirements. This would include factors such as torque requirements, duty cycle, and speed range.

Once the application requirements are defined, the controller kilovolt ampere can be determined and the proper motor selected. The primary concern in sizing the controller kilovolt ampere involves providing sufficient current to produce the required torque. A primary concern in selecting the proper motor involves ensuring that the motor temperature rise does not exceed its insulation rating.

In this chapter first some background information will be given about electric motors in general. After that three types of VFD's will be discussed and some detailed iformation will be given.

9.2 Motor drives

9.2.1 General

In a general sense, one can say that a motor drive is an apparatus that
1. transmits motion or
2. supplies the motive force to a motor in such a way as to control its speed.

These controls may be as simple as an adjustable pulley or as complex as a microprocessor based VFD. Out of the six most common drive methods that Greenberg [Greenberg et al., 1988] mentions, electronic VFD's are becoming increasingly a preffered method thanks to the advances in semiconductor technology.

9.2.2 Types of electronic motor drives

Basically, an electronic motor drive is a device that controls the speed of a motor by varying the magnitude of one of its controllable variables such as *voltage*, *current*, or *frequency*. Of course, the technique used to vary the speed will largely depend on the type of load the drive is going to have, and since these loads have some sort of motor, either AC or DC powered, it is reasonable to make a drive classification based on tthe type of motor. Motor drives can be classified, based on the type of motors, mainly in two categories:
1. D.C. drives;
2. A.C. drives.

These two branche out to more types (see figure 9.1).

DC drives branch into single-phase, three-phase and chopper drives, and so on [Rashid, 1988]. AC drives can be divided into synchronous motor drives, and induction motor drives.

VFD's consist, mainly, of six parts (see figure 9.2):
1. the rectifier or converter circuit;
2. the DC link;
3. the inverter circuit;
4. the control processing circuitry;
5. the control signals from the operator;
6. the control signals from the motor

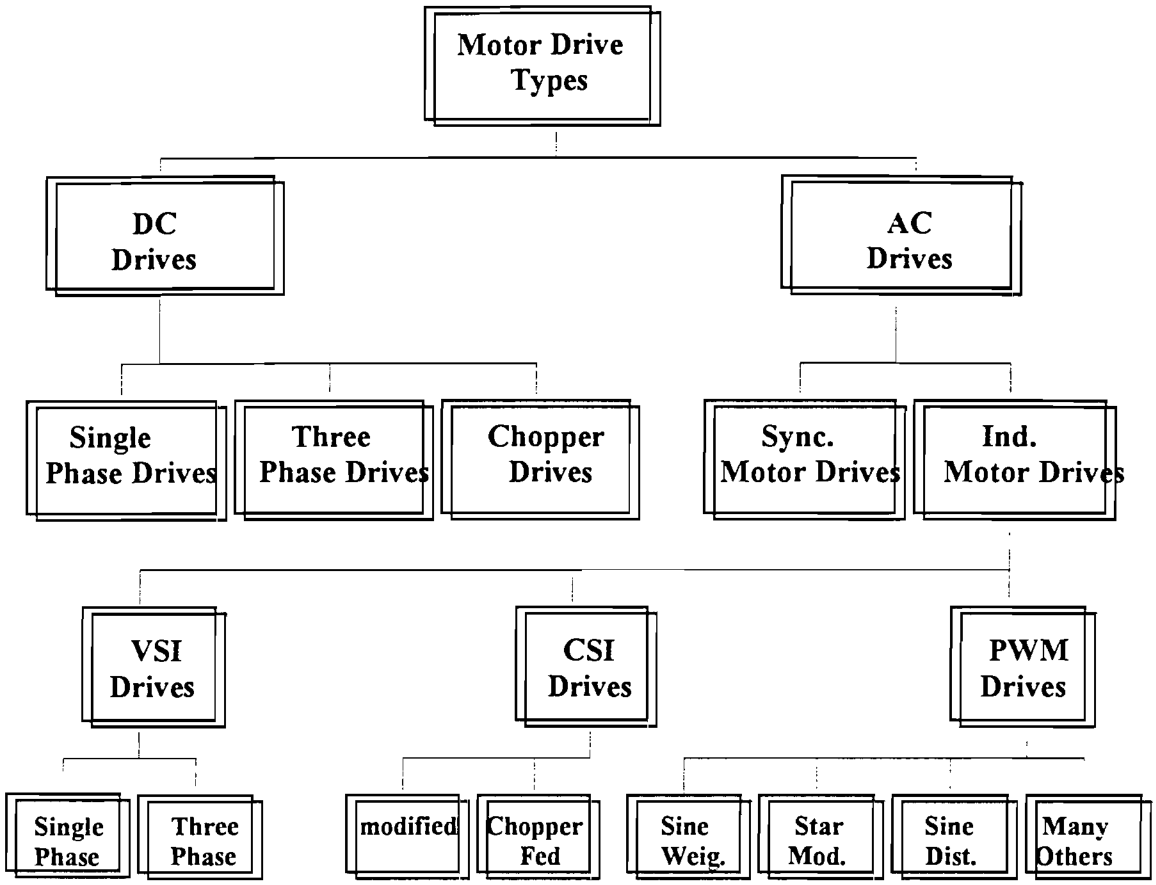


Figure 9.1: **Main motor drive classification including induction motor VFD's.**

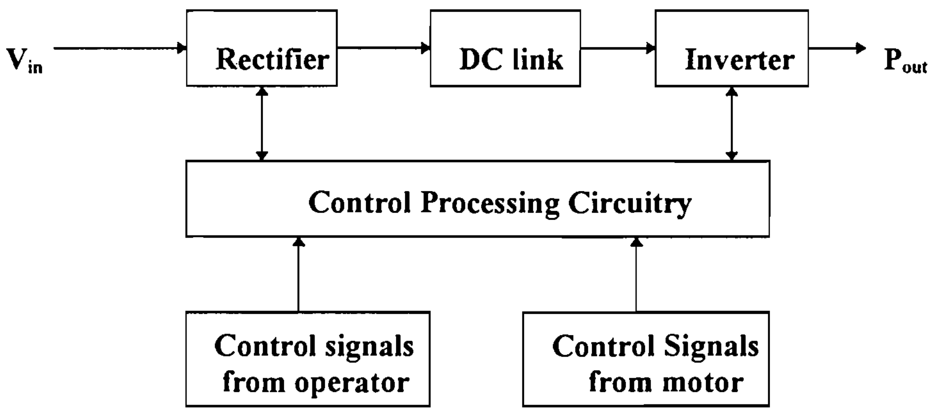


Figure 9.2: **Simplified block diagram VFD**

And since induction motors tend to dominate the market, manufacturers have come up with a great variety of drives to satisfy the industrial as well as the residential consumers. Thus for induction motors, VFD's can be further divided into:
1. Voltage Source Inverter (VSI) drives;
2. Current Source Inverter (CSI) drives;
3. Pulse-Width Modulated (PWM) drives.

These drives are, thanks to new advances in power electronics, which has allowed for the development of a great variety of VFD's, each one of which further divided into more types. For instance, PWM drives into sine-weighted modulation, star modulation, sine distributed, staircase, and many more.

These three drives will be discussed later.

9.3 Adjustable frequency controller characteristics

The speed is a function of the number of poles and the applied frequency as shown:

$$\text{speed} \propto (K \cdot \text{frequency}) / \text{poles}$$

The torque is proportional to flux density in the air gap which is proportional to the Volts/Hertz (V/Hz) applied to the motor:

$$T \propto \phi_{\text{air gap}} \propto V/H$$

From these relations it can be seen that an induction motor will run at variable speed and produce constant torque over its speed range by varying the voltage with the frequency (see figure 9.3). As the frequency is varied, the motor has a particular speed, torque, and current characteristic for each frequency as shown in figure 9.4 for a constant V/Hz output.

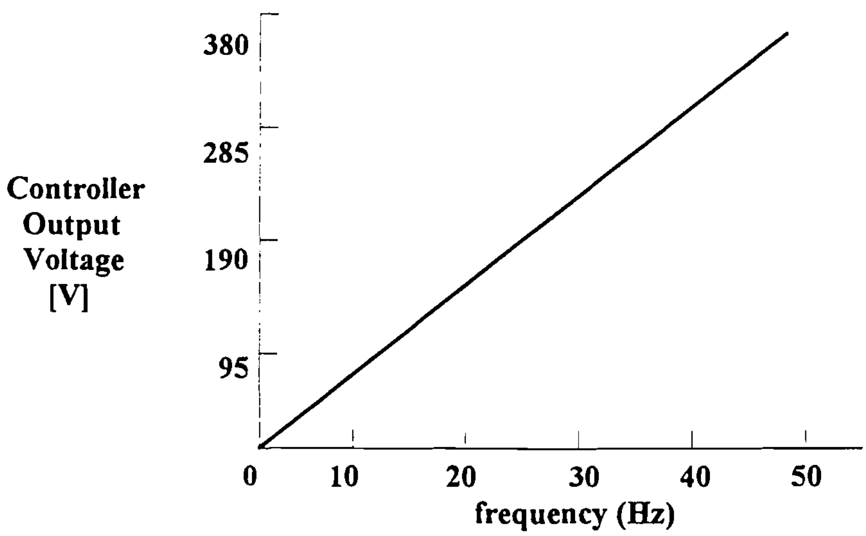


Figure 9.3: V/Hz without offset voltage.

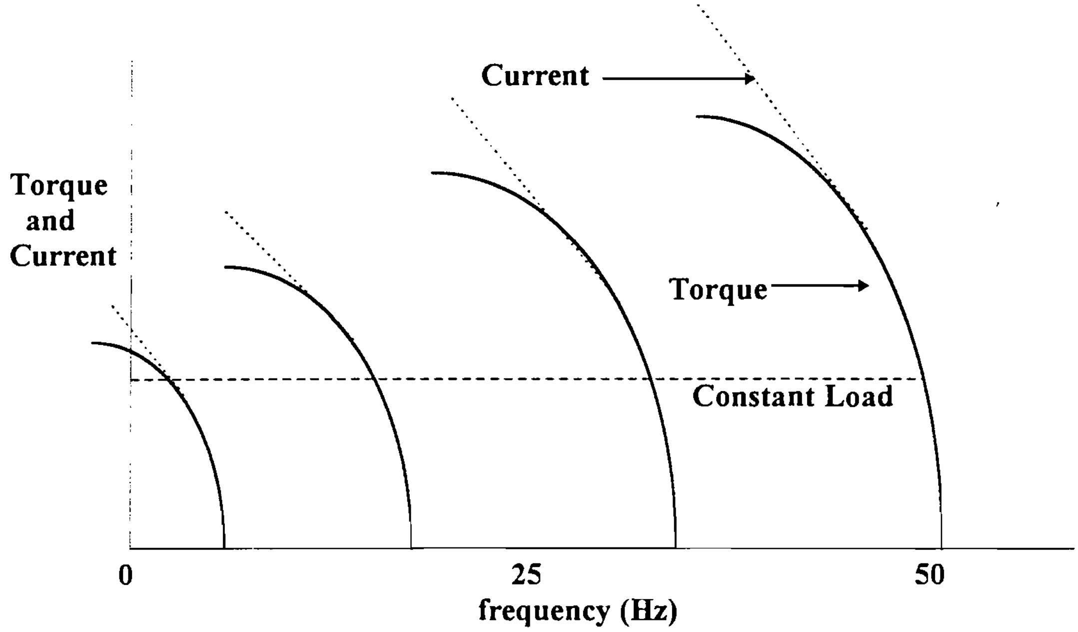


Figure 9.4: Torque and current versus frequency for constant load and no offset value.

As the frequency decreases, the maximum torque available and the breakdown torque decreases. This is caused by the stator resistance voltage drop which becomes significant at low frequencies where low voltage would be applied. This reduction in maximum torque at low frequency can be overcome by introducing an offset or voltage boost at low frequency. This voltage would be adjusted to offset the resistive voltage drop of the stator winding. The resulting V/Hz characteristic is shown in figure 9.5. Once the offset voltage is adjusted, the motor will develop constant torque over the speed range.

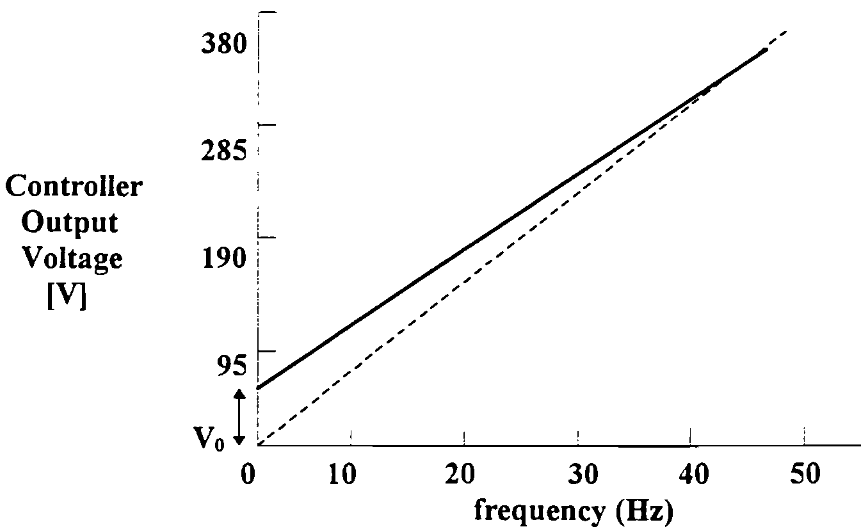


Figure 9.5: V/Hz wirth offset voltage.

It should also be noted that, typically, the torque that the motor developed over the entire speed range is between the breakdown and synchronous speed point. The locked rotor and pullup torque is not encountered when an induction motor is started from rest and run from a controller. The speed-torque and current characteristics shown in figure 9.4 would apply only to full voltage starting.

In addition, the current supplied by the controller is only that amount necessary to meet the torque requirements. Since the controller output generally followa a V/Hz curve, the motor never sees locked-rotor current. This results in a soft start characteristic and is one of the advantages of using an adjustable frequency controller.

Up to this point, only constant torque operation has been discussed. The V/Hz curve, however, can be adjusted to provide constant horsepower operation above synchronous speed. Figure 9.6 shows the resulting speed-torque curve of an induction motor and the associated controller V/Hz output. Constant torque is provided below 50 Hz and constant horsepower above 50 Hz.

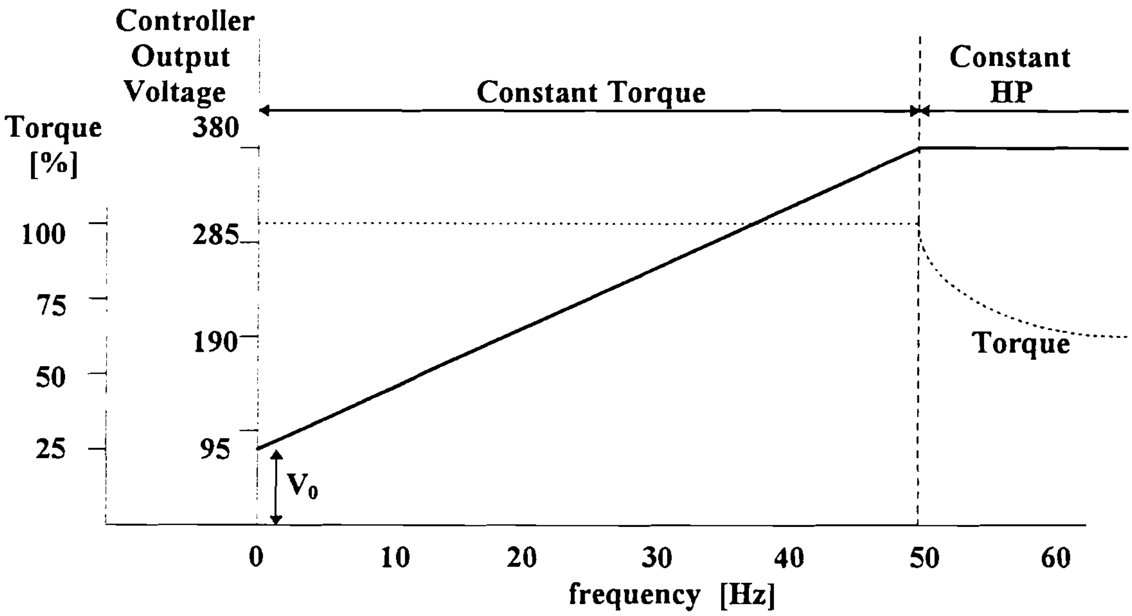


Figure 9.6: Adjustable frequency controller characteristics.

9.4 Types of adjustable frequency controllers

The above discussed theory, mentioned before would generally apply to any of the three types of controllers available:

1. VSI controller;
2. CSI controller;
3. PWM controller.

Now these types will be discussed.

9.4.1 VSI controller

A VSI controller typically utilizes a phase controlled rectifier to generate the required dc voltage level for the desired V/Hz ratio. This voltage is then filtered by a large dc link reactor and capacitor before being inverted to the required frequency for the desired motor speed. The frequency of the output is determined by the switching of the transistors or semiconductor-controlled rectifiers (SCR's) in the inverter section. A block diagram of this approach and the associated voltage is shown in figure 9.7 for the six-step output.

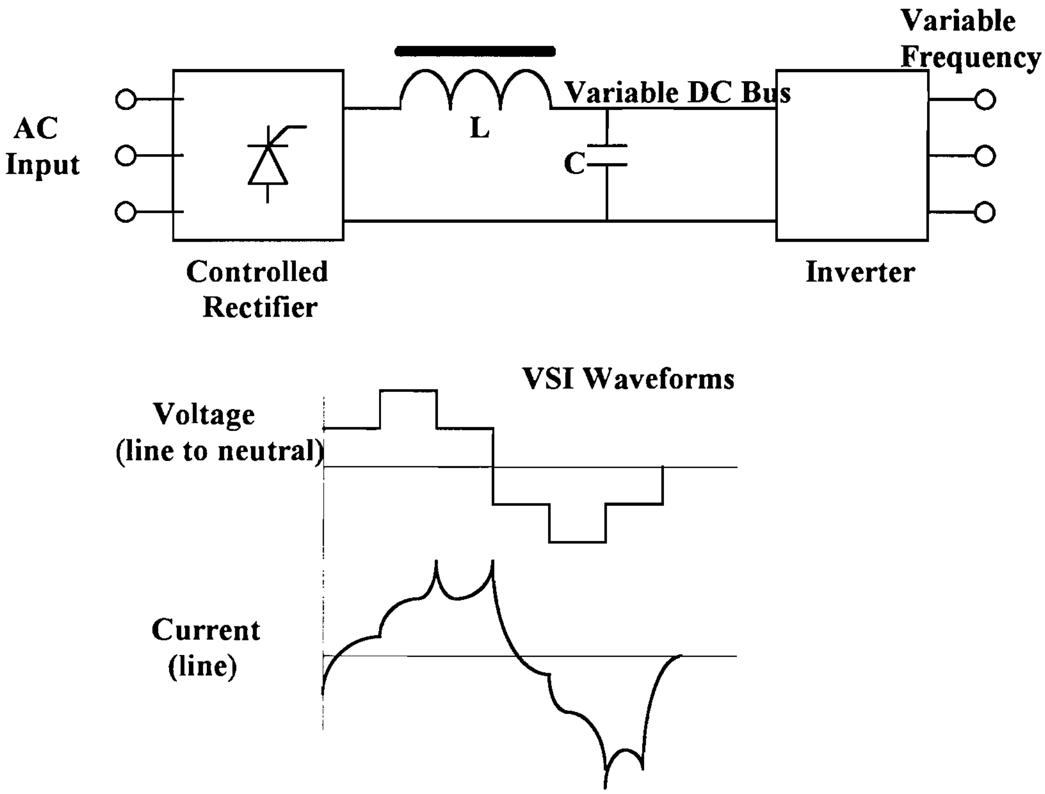


Figure 9.7: Variable source input controller.

This type of controller produces harmonic voltages. The resulting harmonic currents depend on the load impedance at the harmonic frequency. These harmonic currents are limited by the induction motor leakage reactance. Induction motors with higher valuesof leakage reductance will have less harmonic current and lower harmonic losses [Connors et al., 1983].

9.4.2 CSI controller

A CSI controller (see figure 9.8) also typically uses a phase controlled rectifier to generate adjustable dc voltage. The phase controlled rectifier produces the required voltage which is then filtered by a large dc link reactor to constant current. The inverter section then produces the required variable-frequency current, and the motor voltage follows the current.

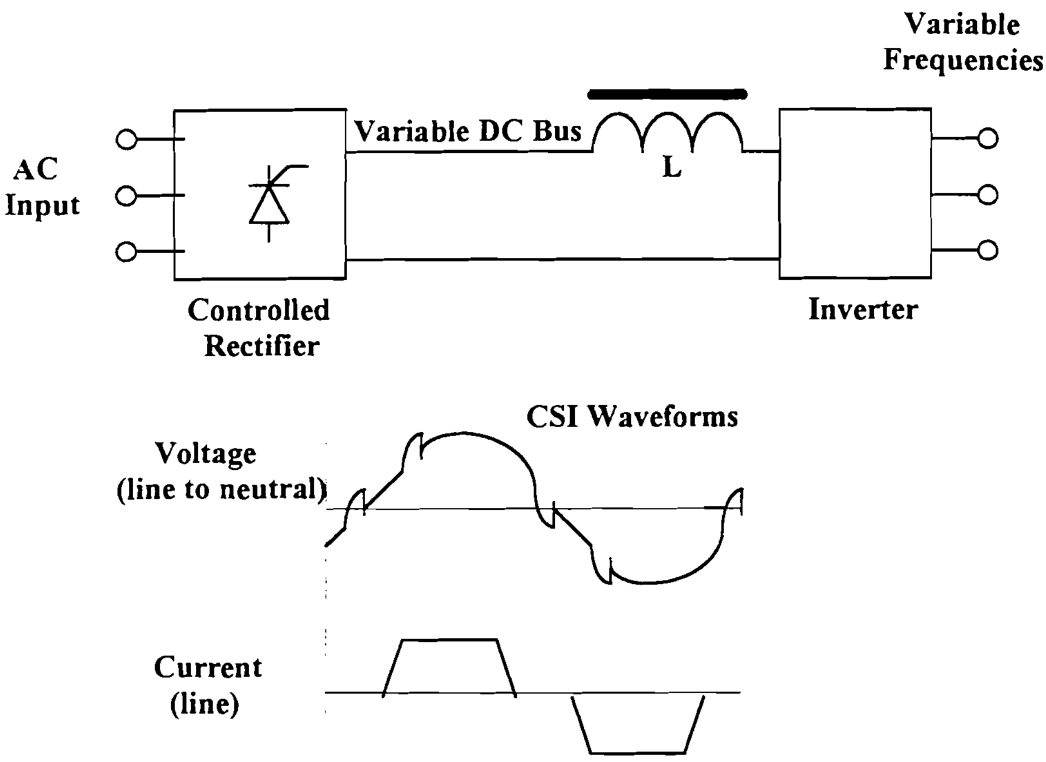


Figure 9.8: Current source input controller.

A typical six-step output is also shown in figure 9.8. This type of controller produces harmonic currents, and the harmonic voltages are limited by the induction motor leakage reactance. In contrast to a VSI controller, though, lower leakage reactance results in reduced harmonic voltages and hence lower losses.

9.4.3 PWM controller

A PWM controller (see figure 9.9) is a controller which uses a fixed diode rectifier with a small filter capacitor to generate constant potential dc. The inverter section is controlled to produce variable voltage and variable frequency as shown in figure 9.9. This technique generally uses higher switching frequencies at lower motor speeds and lower switching frequencies at higher speeds.

This type of controller produces harmonic voltages as is the case of a VSI controller. The harmonic content, however, is dependent on switching frequencies, and is generally much lower than for a VSI. Higher values of leakage reactance for induction motors are desirable for limiting currents.

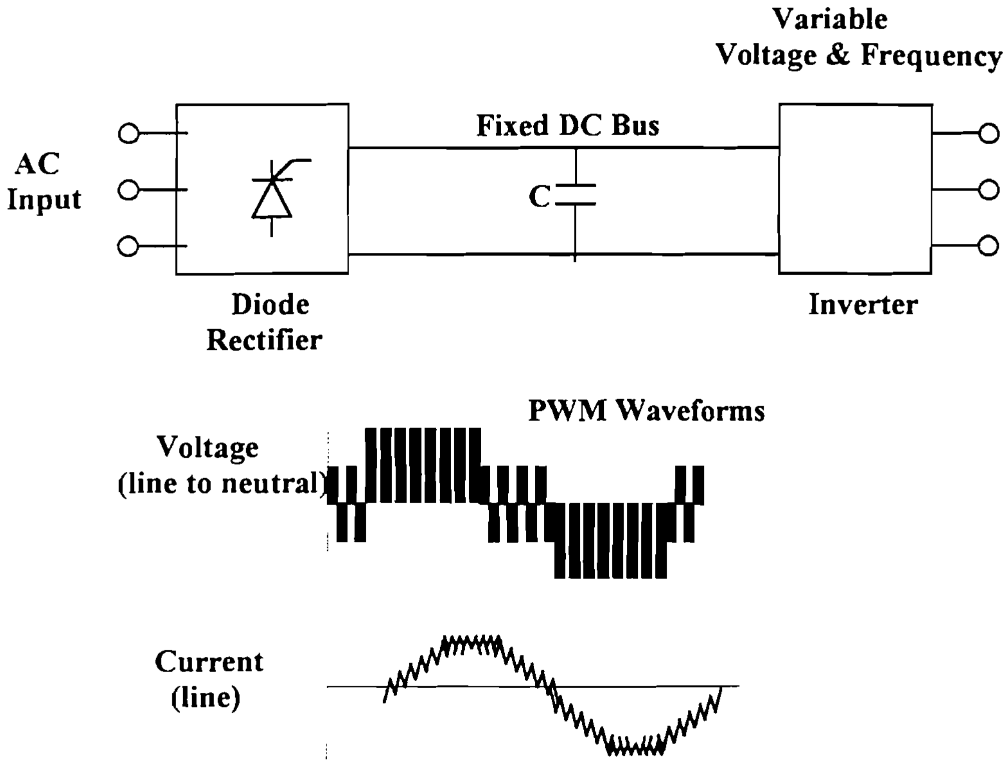


Figure 9.9: PWM inverter.

9.4.4 Table of features of the main VFD's

Table 9.1 gives a compilation of several key VFD features.

Table 9.1: Some features of the main induction motor VFD's.

Feature	VSI	CSI	PWM
Rectifier Device	SCR's	SCR's: hp*	Diodes: lp* SCR's: hp*
DC link Device	LC filter	L filter	LC filter
Inverter Device	Transistors: lp* GTO's, SCR's: hp*	SCR's: hp*	Transistors: lp* GTO's, SCR's: hp*
Torque control	Voltage & Frequency	Current, Voltage & Frequency	Current, Voltage & Frequency
# of Motors	Many	Single	Many
Motor Impedance	Independent of	It is matched	Independent of
Power factor	Low 0,90's	Low 0,90's	High 0,90's
Efficiency	Medium	Low	High
Other	Hi-output frequencies	Simple Controls	Low-magnitude harmonics

hp* = high power applications, & lp* = low power applications

9.5 Application requirements

9.5.1 General

Definition of the application is the key to successful operation of a VFD. This information is necessary to determine the kVA rating of the controller and the correct motor for the application. As the pumps at AFS are centrifugal pumps, the variable torque application will now be discussed.

9.5.2 Variable torque applications

Variable torque applications, such as centrifugal pumps, are by far the easiest to work with [Connors et al., 1983]. The torque varies likewise the square of speed as shown in figure 9.10. Since the starting torque requirements are very low, very little, if any, offset voltage is required to start the drive. This is valid for all pumping applications as long as the fluid being pumped has a minimal solids content.

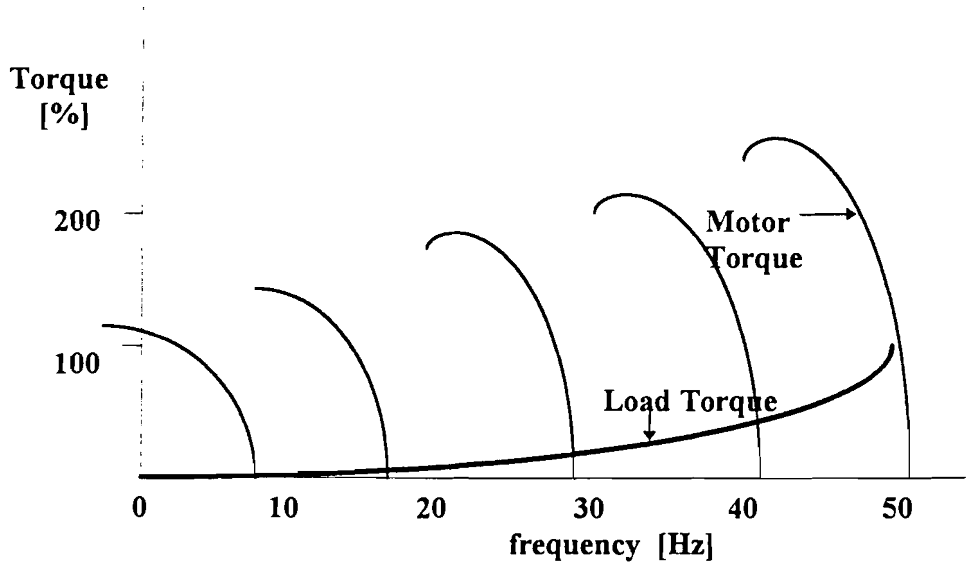


Figure 9.10: Torque versus frequency for variable torque load and no offset voltage.

For centrifugal loads, the torque requirements vary with the square of the speed, and the power requirements vary with the cube of the speed. Consequently, the maximum load would generally occur at full speed, and the required input controllers kilovolt ampere would then be:

$$kilovolt-ampere = \frac{\sqrt{3}\cdot(line-to-line-voltage)\cdot(ratedcurrent)\cdot 1{,}1}{1000} \qquad (9.5.2.1)$$

The rms sine wave current is increased by 10 percent to take into account that the current from the input controllers is nonsinusoidal. Using the data from the different motors at Rijk the required kilovolt amperes are obtained (see table 9.2).

Table 9.2: The required kilovolt amperes for the different motors at Rijk.

Pump	Motortype	kilovolt Ampere
101 - 104	Hemaf 4H 315 SWT 20	141,2
105 and 106	Loher A 315 SA-2	143,4
107 - 110	Loher EMGV-315LB-0	142,6
111 and 112	Loher EMGV-315MD-	151,3
201 - 203	Loher DNSW-315MB-	155,6

9.5.3 Motor application

The primary concern with operating standard induction motors on adjustable frequency controller power is the additional temperature rise over sine wave temperature rise. This additional heating is caused by the harmonic content of the input controller and the reduced cooling at speeds lower than full load speed [Connors et al., 1983].

Limiting the additional induction motor temperature rise on adjustable controller power is essentially a matter of reducing the harmonic related losses. Thus better performance on VSI or PWM input controllers can be obtained by designing the required horsepower rating in a larger frame size. This provides additional material for heat dissipation and allows increased leakage reactance, which limits harmonic losses, to be designed into the motor. Using standard overframed motors on VSI or PWM input controllers would not generally be recommended because leakage reactance decreases with increasing horsepower for standard motor designs, and consequently, harmonic losses would increase. Better performance can be obtained on a CSI controller, however, by using a larger horsepower standard motor design. The lower leakage reactance would limit the harmonic losses associated with the CSI controller output. A CSI, however, would not lend itself well to operating smaller horsepower ratings, whereas a VSI or PWM would be able to operate smaller horsepower ratings.

For any given motor frame size, the losses will vary depending on the NEMA speed torque characteristics. The various speed torque characteristics are associated with different induction motor impedance characteristics which can affect losses. In general, Design B characteristics would be more desirable for VSI and PWM controllers, and Design A characteristics would be more desirable for CSI controllers. NEMA Design C and D motors would not be recommended for use on any adjustable frequency controller.

9.6 Benefits of the VFD's

The main benefit from VFD's is their energy efficiency.

There are other benefits in using adjustable speed not directly connected with saving energy. The dollar value of these additional benefits is hard to fix and is made mainly by user evaluation in a specific application. A summary of main additional benfits follows [Hickok, 1983]:

1. **Soft start of drive**. VFD's inherently start with low inrush currents. If the driven load has an exponential speed torque curve for starting, this is only 30%-50% of full load current. If it requires constant torque this may be as high as 100%-150% full load current. Fixed speed drives and those driven by hydraulic couplings may have 400%-650% inrush current. With the exception of non-electrical type drives such as steam and gas turbines this has the following advantages:
    - *Power system reduced.* Because of high inrush currents, power systems must be designed to hold their voltage up during starting. This is particularly important if there is one drive much larger than the rest on a given utilization bus. To accomplish this with motor high starting inrush currents, the power system must be stiffened up at times with more than sufficient transformer capacity. This results also in higher than otherwise needed switchgear interrupting duty. Use of VFD's can sometimes save money on the power system by using a system with lower starting capability. They are particularly suitable for starting on weak power systems, and at times, as in remote pump stations, this can be the major reason for their use. Note in passing that as VFD's do not provide a short circuit contribution to other faults on the system this can be a further help towards reducing interrupting duty on the bus. The only exception to this is when in some types of inverters the drive happens to be operating in a regenerating mode at the instant of fault.
    - *Rotating equipment life lengthened.* Motor life is determined in part by the number of starts it makes. Normally high locked currents and the accompanying magnetic forces involved in fast and uncontrolled acceleration times causes high temperatures, end turn coil movement and uneven thermal expansion of rotors. Ultimately the thermal and magnetic forces hasten failure. In larger motors there is a limitation of only two starts per hour. In VFD's with low starting currents, acceleration time is controlled and is done very slowly on a programmed basis. There are no limitations on the number of starts per hour, and life is made much easier for the motor. This can well extend rotor life over that of a fixed speed motor. Lack of high impact and transient torques also tends to increase the life of couplings and gear teeth in gears.
2. **Increased motor life**. VFD's operate a good part of their life at something less than rated speed and power. As speed drops below rated value, bearing life tends to go up. This is particularly true in small drives with ball bearings where bearing life goes up by nearly the square as speed drops. A drive operating at less than rated horsepower has its insulation temperature reduced and insulation life is approximately doubled for each 10° drop in temperature. The d.c. link in nearly all kinds of adjustable frequency and d.c. drives along with the isolation transformer isolate and shield the motor from power system problems.

This may include low incoming voltage which is adjusted for in the control circuit. It can correct for unbalanced a.c. voltage on the power system which causes negative sequence torques and thus additional motor heating.

3. **Process ride through**. Fixed speed drives operating when a power system disturbance such as a nearby fault occurs, cannot continue operation for more than a few cycles with no voltage. When voltage returns after the interruption, the drive has slowed to the point where it cannot reaccelerate the driven load unless the load is reduced immediately as with automatic unloader valves on a compressor. Very few drives have these, and thus most drives will be tripped off by locked rotor protection, when motors stall trying to reaccelerate at reduced torque while fully loaded. This is indicated in figure 9.11.

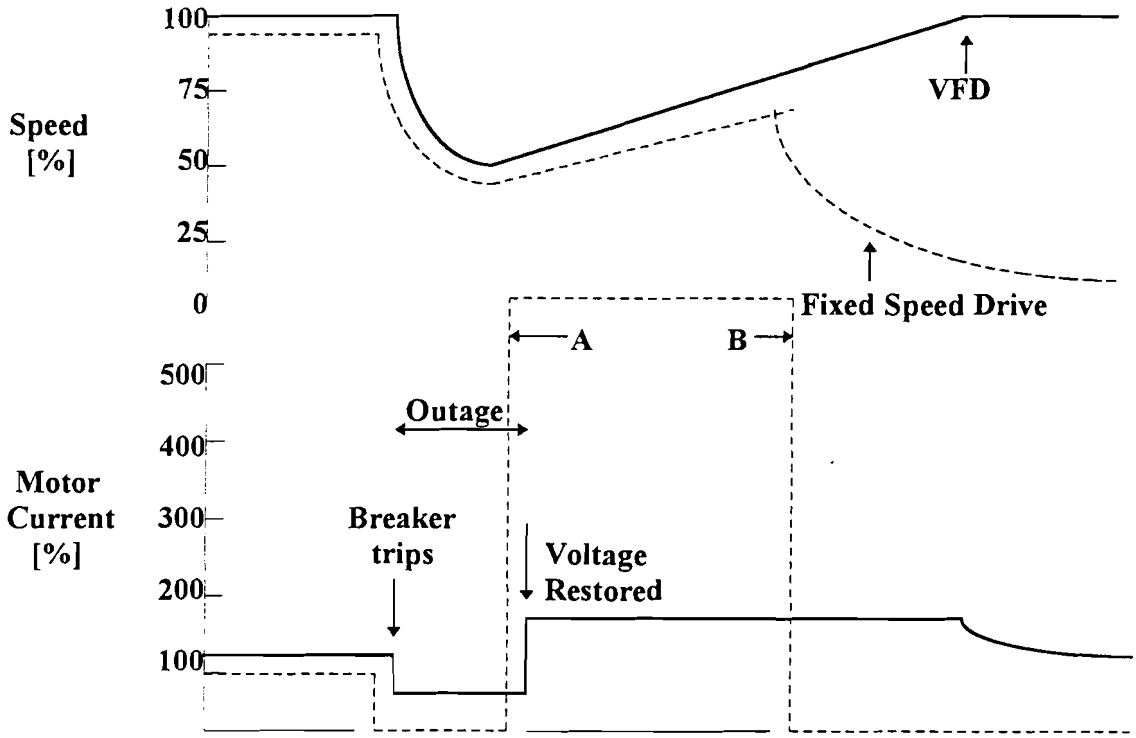


Figure 9.11: **Process ride through a power outage for fixed speed and VFD's [reprinted from Hickok, 1983].**

A: Fixed speed drive attemps reacceleration.

B: Rotor thermal protection trips breaker.

By contrast an adjustable frequency or d.c. drive has the capability to catch the drive when it is slowing down and under slow reacceleration with full load torque available bring the drive back up to its preset speed. This ride through capability can save many process shutdowns from unexpected voltage disturbances on the power system.

4. **Increased speed range**. VFD's do not know that 50 Hz is the standard frequency. Inverters can take their motors to well over their 50 Hz speed if needed. In some drives additional speed over rated is sometimes desired, and where horsepower rating is not exceeded, the capability to extend the speed range is available. For practical limits this may be only 10% to 15% more. Torsional problems on the motor and/or drive equipment may occur somewhere beyond that point. Since horsepower in many cases goes up as the cube of the speed it won't take long before a horsepower limit is reached.
5. **Extending operating range before surge line**. As illustrated in figure 9.12 a fixed speed pump as it is throttled back to reduce flow, can only be reduced in flow so far before it encounters the surge line, a point below which operation is unstable and possible damage to bearings and seals is likely to occur. Because of control system drift or error or change in the line usually a 10% margin of safety is held away from the line. As pressure is reduced for a given flow, the available control range is inherently increased as shown in the set of curves. The surge line goes down and to the left. Adjustable speed provides greater flow at less pressure and therefore a greater possible control range before the surge line is encountered. The fixed speed pump given in figure 9.12 would have a usable operating range of 75% to 100% or only 25% range in flow. An adjustable speed pump would be able to utilize 30% to 100% or 70% flow range.

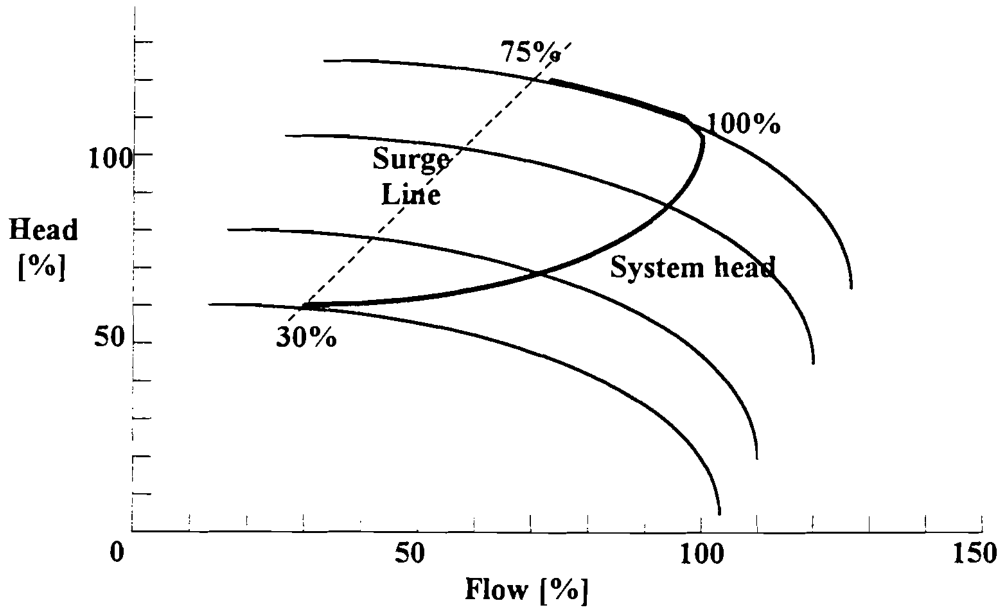


Figure 9.12: The flow range in pumps is increased by adjustable speed over fixed speed.

However, from all of these benefits, the one most recorded in the literature is *energy savings*. The potential energy savings of using VFD's by the year 2000 may be nearly 170 billion kWh [Domijan et al., 1992]. Their energy efficiency makes them very appealing to industry, and since the object of most industrial purchases is to minimize the long run cost to the company, VFD sales should continue to show strong growth.

9.7 Conclusions

VFD's can be used successfully for many applications once the drive requirements are defined. The adjustable frequency controller replaces a large source of current and as such is current limited rather than horsepower limited. For drives other than those represented by most centrifugal loads, application details are essential to determine the proper hardware. Both the adjustable frequency controller and motor must be evaluated for a particular requirement. Once the application requirements are defined and the combined controller and motor characteristics understood, successful operation of the drive can be assured.

Chapter 10

Power quality for VFD's

10.1 General

Each of the three VFD drives mentioned in chapter nine impacts the power quality, power factor and harmonic content as a function of the line (source) inductances. The first two will be discussed in this chapter, the harmonics in the next chapter.

First the process commutation notching will be dealt with. After this electrical noise, the power factor and grounding will be discussed.

10.2 Commutation notching

Commutation is the process by which one set of thyristors (or diodes) turns off and the next set turn on. Commutation notching occurs when thyristors are used.

In a full six-pulse converter, the thyristors operate in pairs to convert ac to dc by switching the load current among the six thyristor pairs six times per ac cycle. During this process, the current begins to transfer from one phase to the next creating a momentary phase-to-phase short circuit. Source inductive reactance prevents instantaneous transfer (commutation) resulting in a commutation notch.

The duration of this short circuit (notch width = μ) is a function of the total system inductance and the dc output current [IEEE standard 519, 1992].

Figure 10.1 defines notch width and depth. The notch depth and area will differ depending where in the system it is measured. For example, the notch depth at point *A* in figure 10.2 will be 100%. At point *B*, it is calculated as follows [Jarc et al., 1985]:

$$NotchDepth(\%) = \left(\frac{L_1 + L_2}{L_1 + L_2 + L_3}\right) \cdot 100 \qquad (10.2.1)$$

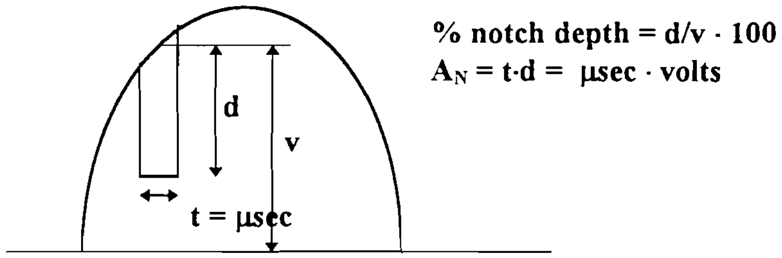


Figure 10.1: Definition of notch depth and notch area.

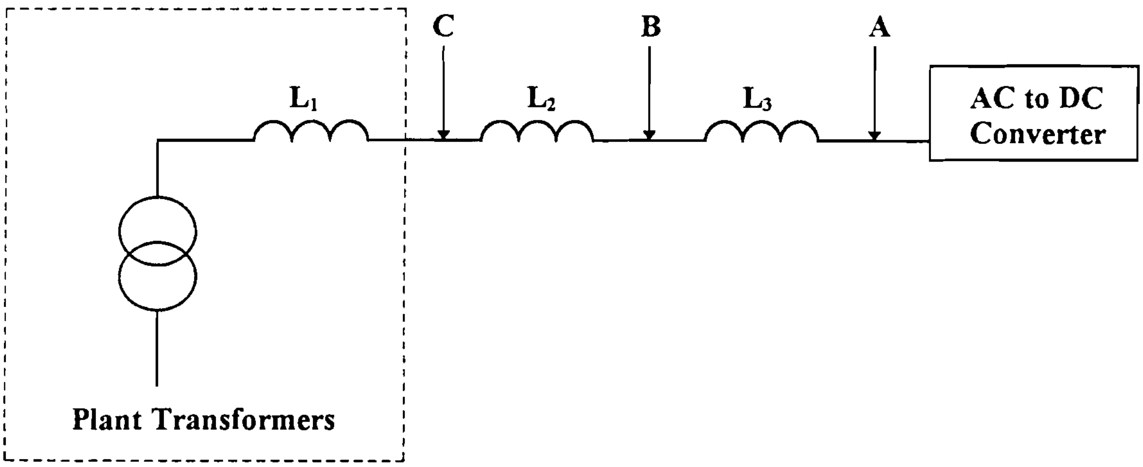


Figure 10.2: AC line impedance distribution.

Since a line notch is a sudden change in voltage (dv/dt), resistor/capacitor *(snubber)* networks will begin to discharge/charge during commutation notches. Where multiple drives are applied on a common bus, the composite commutation notching can overwork these snubber circuits and in severe cases, fail catastrophically. These networks are applied to absorb transient voltages occurring across the thyristors due to high speed electronic switching (Ldi/dt). If these networks are out of tolerance or nonfunctional, a transient voltage overshoot may occur with a high incidence of thyristor fuse blowing and/or drive component failures.

Also other sensitive equipment fed from the common system can exhibit power quality/control problems. A practical upper limit: The repetitive peak deviations of the fundamental line voltage from the instantaneous value of the line voltage may not exceed 25% of the crest working line voltage [Jarc et al., 1985].

These line notches are also rich in high frequency harmonics and are propagated throughout the power system. Sensitive electronic equipment such as computers, PLC's and instrumentation is especially sensitive to the high frequency content due to line notching. For this reason, it is always a good idea to keep these sensitive type loads electrically isolated from these drives.

Line notching can also cause thyristor misfiring. This can happen when notch width exceeds gate pulse width or with excessive notch depth. As a recommended practice, VFD's should be selected /designed to operate with maximum voltage notch widths of 250ms and depths of 70% of rated peak line current [IEEE standard 519, 1992].

Table 10.1 is excerpted from [IEEE standard 519, 1992] that defines commutation notching limits for low voltage systems. For those systems which may have notching out of limits, adding commutation reactance ($L_1 + L_2 + L_3$ in figure 10.2) is the most practicable solution. Generally this is best accomplished by making $L_2 + L_3 >> L_1$ [Jarc et al., 1985].

Table 10.1: Low-voltage system classification and distortion limits.

	Special Applications •	**General System**	**Dedicated System †**
Notch Depth	10%	20%	50%
THD (Voltage)	3%	5%	10%
Notch Area (A_N)‡	16.400	22.800	36.500

Note: The value A_N for other than 480V systems should be multiplied by V/480

• Special Applications include hospitals and **airports**

† A dedicated system is exclusively dedicated to the converter load

‡ In volt-microseconds at rated voltage and current

The addition of a series commutation reactor in the drive itself or a drive isolation transformer easily serves this purpose. Another benefit of adding commutating reactance is that the current harmonics generated can be significantly reduced. However, a dilemma can occur when the addition of inductance to decrease notch depth results in too large of a notch area. Reducing L_1 (larger stepdown transformer) or the addition of power factor capacitors at point *C* in figure .2 amy be alternatives but effect other system design parameters (short circuit levels and harmonics) [Shipp et al., 1996].

10.3 Electrical noise

Higher frequency harmonics caused by commutation notching, thyristor induced transients (Ldi/dt), high frequency (radiated or conducted) components caused by high frequency switching in the inverter section (primarily CSI drives), inadequate or improper signal grounding, electro-static discharge from insulation to ground (common mode noise) as well as use of walkie-talkies in close proximity to drives, have all been known to cause drive malfunctions. The higher frequency sources and thyristor induced transients are sometimes called *crosstalk* or electrical noise.

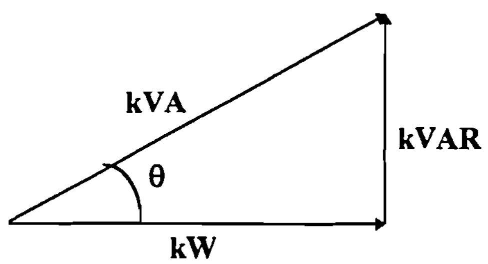
If this electrical noise is coupled into the controls or signal paths (such as thyristor gate leads), thyristor misfiring, unstable speed control, erratic behavior, and/or control board failures can and do occur. This is especially a concern where multiples drives are fed from a common bus without individual commutation reactors or isolation transformers applied. The application of newer digital VFD's to existing analog drive systems may result in similar problems. Solutions could consist of adding commutation reactance, isolation transformers, and carefully solving grounding installation problems. Where digital and analog drives are intermixed, adding MOV's and/or changing out thyristors to be more compatible with each other, are additional considerations [Shipp et al., 1996].

10.4 Power factor

10.4.1 General

The adjustable frequency controllers have different power factor characteristics than a fixed speed induction motor. In addition, each type of adjustable frequency controller has its own specific power factor characteristics.

It is of interest to address the issue of power factor because utilities generally have not only a kWh charge, but also a demand charge. This demand charge is directly related to the power factor. The distribution system size is based on kVA. The relationship of kVA to kW and power factor is shown in figure 10.3.



Power factor = $\cos\theta$ = kW / kVA

Figure 10.3: Power factor on sine wave power.

As can been seen from figure 10.3, a lower power factor (larger θ) causes an increase in kVA to produce the same kW.

Since induction motors and adjustable frequency controllers are inductive loads, they operate with a lagging power factor. The power factor for adjustable frequency controllers also follows this relationship, but the controller input current is no longer a sine wave. The current waveform is closer to a square wave then it is to a sine wave [Jarc et al., 1985].

Most adjustable frequency controller manufacturers define power factor as displacement power factor. It is defined as the angle between the fundamental line-to-neutral voltage and the fundamental current. It cannot be used to determine total kVA, but rather only the fundamental kVA. The total kVA is the fundamental kVA plus the kVA due to the harmonics of the nonsinusoidal input current waveform. This is graphically shown in figure 10.4.

Typically, total power factor is defined as the relationship between the kW and kVA. For sine wave power, total power factor equals displacement power factor. On adjustable frequency controllers, however, the total power factor is not equal to the displacement power factor. This again is due to the differences in harmonics. The total power factor will always be less than the displacement power factor.

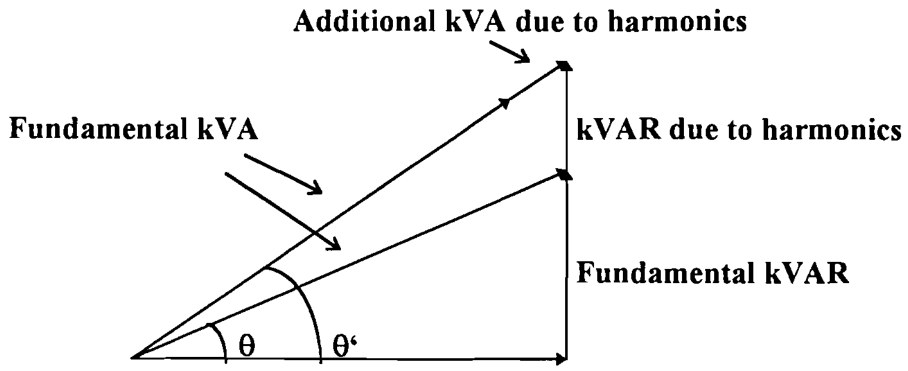


Figure 10.4: Effect of harmonics on power factor.

Previously, two types of demand charges were discussed. When these two methods are used on adjustable frequency controllers, they produce completely different results.

- *power factor method,* which measures the angle θ between the voltage and current. This measurement device typically only measures fundamental curent and, therefore, measures displacement power factor.
- *kVA demand method,* which measures the maximum kVA for a certain amount of time. This maximum kVA measurement includes the harmonics and, therefore,measures total power factor

Thus the demand charge will be dependent upon the type of measurement used.

10.4.2 Power factor correction

It is frequently desirable to correct for power factor in order to lower the amount of kVA required to do a certain amount of useful work (kW). A power factor correction capacitor reduces the kVA demand by supplying the reactive current instead of the distribution system. Figure 10.5 shows how this accomplished. Graphically, the capacitor substracts from the magnetizing component (kVAR). Therefore, in order to size the capacitor correctly, the amount of KVAR's which have to be compensated for must be known.

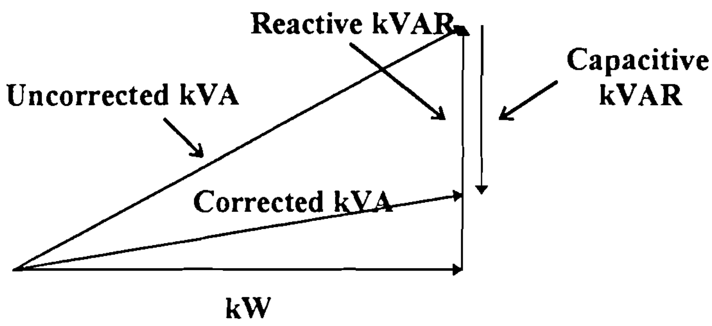


Figure 10.5: Effect of power factor correction capacitor on kVA.

10.4.3 Centrifugal loads

Figure 10.6 shows the characteristics of a centrifugal load which would be characteristic of a centrifugal pump. Since the load is well defined, a general procedure for determining a power factor correction capacitor can be provided.

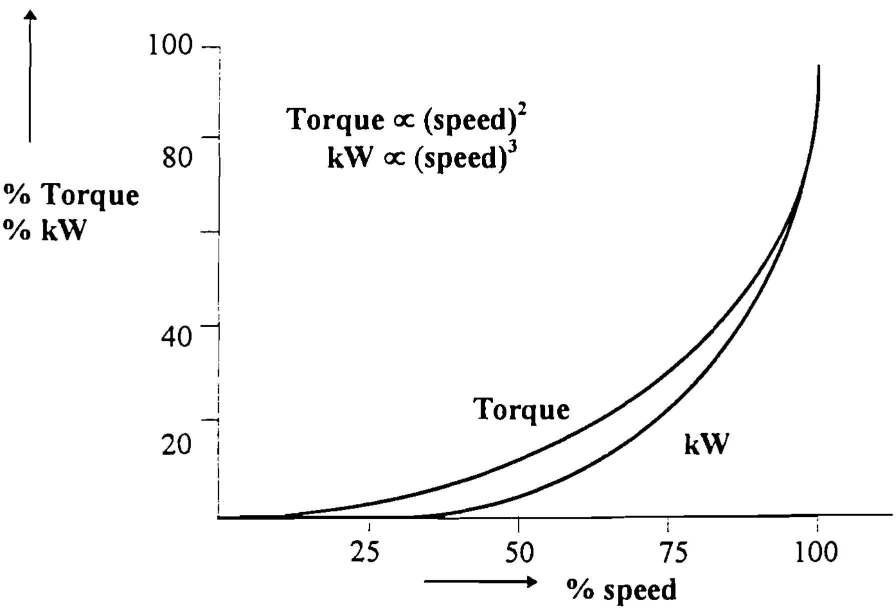


Figure 10.6: Percent kW and percent torque versus speed profile for centrifugal load.

Because the torque varies with the square of the speed, the actual kW requirement can be calculated for any operating point. The input kW and input kVAR requirements can be calculated by determining the input power factor and the efficiency of the drive system. In order to outline a general method of calculating required kVAR, a drive efficiency of 100 percent will be assumed. The resulting relationships would be:

$$\text{input kVAR} = \tan\left[\cos^{-1}(\text{PF})\right] \cdot \text{kW input} \tag{10.4.3.1}$$

where PF is the power factor, and

$$\text{kW input} = (\text{kW output required}) \;/\; (\text{efficiency}) \tag{10.4.3.2}$$

Then, by assuming an efficiency of 100 percent:

$$\text{input kVAR} = \tan\left[\cos^{-1}(\text{PF})\right] \cdot \text{kW output required} \tag{10.4.3.3}$$

The kVAR decreases rapidly as the speed is reduced. This is due to the fact that the kW output required decreases as the cube of speed. This will be the main reference for selecting power factor correction capacitors for VSI controllers [Jarc et al., 1985].

The PWM has a displacement power factor close to unity so the required kVAR would be close to zero. Therefore, displacement power factor correction would not be done on that type of controller. The total power factor can be much lower and the power factor correction would be totally dependent upon the power distribution characteristics [Jarc et al., 1985].

Operating VFD's with thyristor converters at slow speeds generally results in poor power factor. If the number of ac drives is small compared to overall plant load, then the total power factor may not be cause for concern. However, large systems of ac VFD's would require reactive compensation. One characteristic of PWM VFD's is a good power factor (due to the diode front end). Where many smaller drives are applied with expected low speed operation, PWM drives will significantly improve power factor and could reduce the cost of harmonic filtered capacitor banks [Shipp et al., 1996].

10.4.4 Power factor measurements at RijkI and RijkII

In this paragraph some figures for the power factor will be given. These values are important, because next year the energy company will oblige AFS to have a power factor of 0,85 or more. If AFS can't reach this power factor every moment of the year, extra costs must be paid.

In figure 10.7 the power factor in relation to the time has been given. The power factor is calculated by the kW and kVA measurements (see figure 10.8). They took place every 80 msec. for a period of 15 minutes and the average is given.

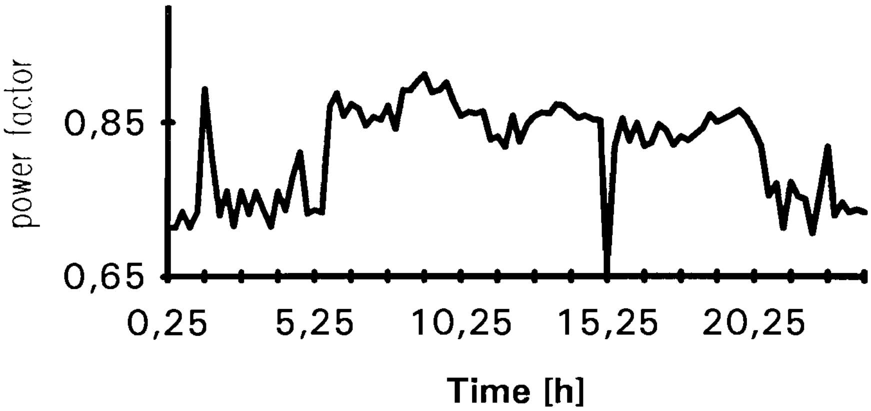


Figure 10.7: Power factor measurements every 15 minutes at RijkI.

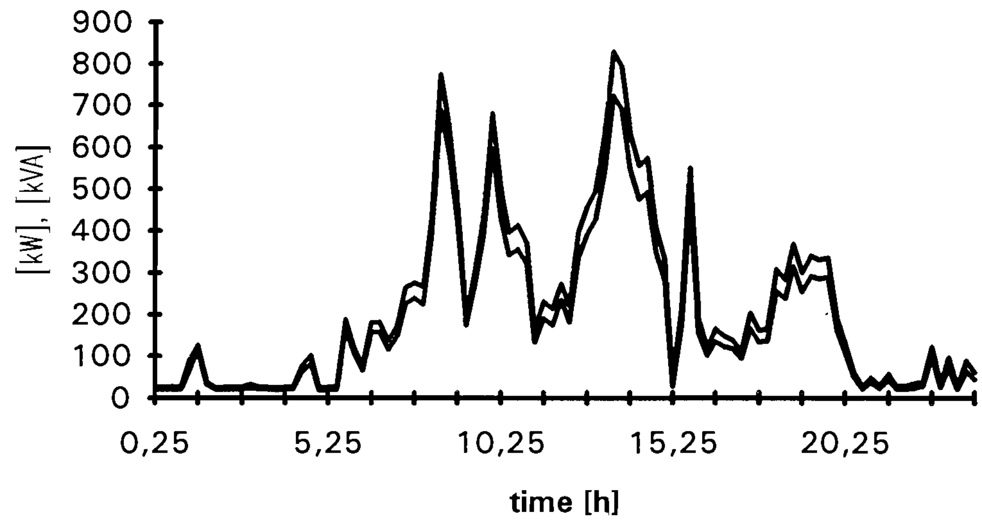


Figure 10.8: The kW and kVA measurements every 15 minutes at RijkI.

It is noticed that the power factor in the morning and evening hours is about 0,75. This is not so good. Because there are also low flowrates at these hours, it can be concluded that by a low flowrate or no flowrate at all, the result will be a bad power factor. The VFD improves the power factor, what is also an important aspect for future considerations.

In figure 10.9 and figure 10.10 the power factor, kW and kVA measurements for RijkII over one day are given. The power factor here is a lot better than that of RijkI.

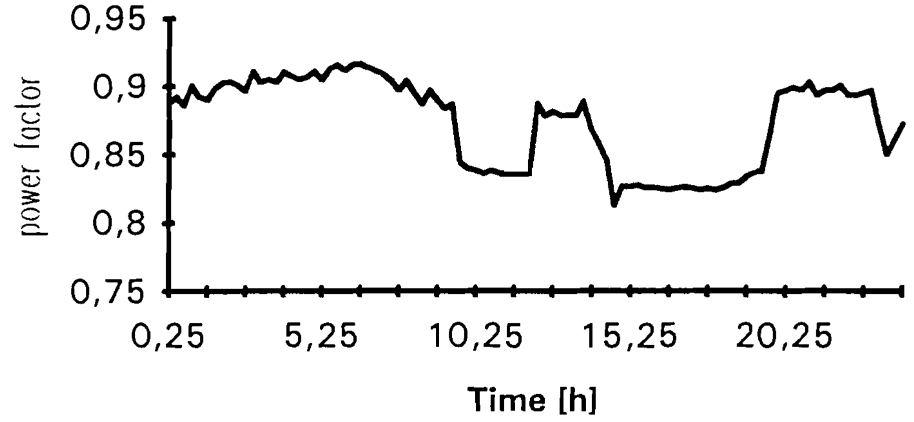


Figure 10.9: Power factor measurements every 15 minutes at RijkII.

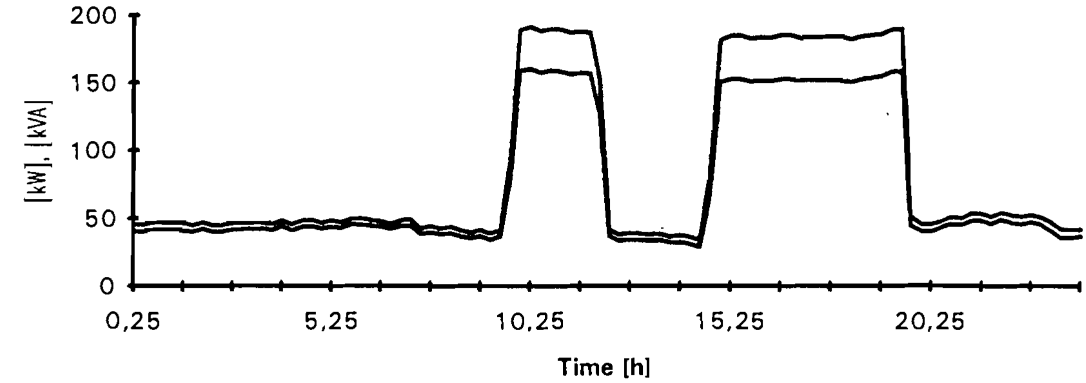


Figure 10.10: The kW and kVA measurements every 15 minutes at RijkII.

In 1998 every energy facility should produce a power factor bigger than 0,85. If not, one should pay extra. From the figures 10.7 to 10.10 one should think that AFS has an average power factor lower than 0,85. This is not true, because the energy company measures the average power factor over one month. It is calculated as follows:

$$\frac{TotalamountkWh}{0{,}85} = TheoreticalkVA \qquad (10.4.4.1)$$

If the theoretical amount kVA (calculated by (14.4.4.1) is:

< the actual amount of kVA absorbed, then: no extra costs;

> the actual amount of kVA absorbed, then there are extra costs;

An example of March 1997 at RijkI is given below:

Actual amount of kWh absorbed: 137637,

so the theoretical kVA = 161925.

Actual amount of kVA absorbed: 160683

So AFS didn't have to pay extra. The last two years AFS always had a larger power factor (average about 0,86) than required. An implementation of a VFD is always good, because it increases the power factor.

10.5 Grounding

There are two areas of grounding concerns for drives or drive systems:

1. power system grounding;
2. signal/equipment grounding.

The majority of the smaller hp drives are applied at 480V. Most 480V systems on which they are applied have a solidly grounded wye-connected source transformer. In many cases, the solid grounding is a National Electric Code (NEC) requirement whether or not VFD's are applied. Consider that if a ground fault occurs internal to the drive itself, the very high magnitude ground fault current available on a solidly grounded system could cause catastrophic failure. If this is a concern, a drive isolation transformer will derive a new ground system for the drive. These isolation transformers also add commutating reactance (reduces harmonics, commutating notching, etc.) as well as a special ground shield between the primary and secondary windings. This shield also gives extra isolation for noise, but costs extra.

Chapter 11

Harmonic distortions

11.1 Introduction

Power system engineers have been dealing with outages, voltage drop, power factor, spikes and surges associated with transient conditions since the invention of the first ac distribution system back in the early 1990's. Adverse power system harmonics caused by the installation of nonlinear type loads on the distribution system has become an increasing concern to utilities in the 1970's and 1980's. With the advances in power electronic converters, more and more of the proportion of motor loads are being applied using power electronic motor drives which use phase controlled rectifiers to convert ac to dc. As mentioned before, the potential energy savings by the year 2000 resulting from the use and operation of the power electronic motor drives could reach as high as 170×10^9 kWh/yr. The main benefit of using power electronic drives is in the economics and energy reduction savings which is typically in the 20% to 50% range over that of conventional motor electro-mechanical drive applications. Other typical benefits include smooth ramp-up and ramp-down acceleration transitions, optimized mechanical load characteristics, equipment reliability, reduced mechanical related maintenance, increased product qualities, reduced audible noise levels, and reduced space requirements. However, there are a few disadvantages associated with power electronic motor drives, but perhaps the single greatest disadvantage comes for the rectifier interface to the utility power system [Phipps et al., 1994].

11.2 Types of power electronic converters

There a just a few basic types of electrical motors which are used extensively in industry. These include, dc motors, ac induction motors and ac synchronous motors. These motor drives share two things in common:

1. they drive electric motors and save on energy costs by optimization of speed and torque;
2. they use 6-pulse phase controlled, full bridge rectifiers which produce distorted voltage and current waveforms in the ac power sytem.

The utility system ideally consists of a constant voltage and frequency (50Hz) source of electrical energy. The power electronic motor drives convert three-phase ac power to variable voltage dc power with 6-pulse type controlled rectifiers. It is this conversion and rectification of electric power which produces undesirable harmonic current and voltage distortion on the ac power system. When the harmonic currents flow in the power system, they can cause problems such as voltage distortion, power factor correction capacitor parallel resonances, power transformer heating and overload, electric meter errors, power cable failures, telephone and communication line interference, and reduced motor efficiency by the additional motor stator and rotor heating and pulsating mechanical torques resulting from the induced negative sequence voltage harmonics. The electric utilities are finding that the total number of large and small rectifier/converter systems which are connected to their systems is significant, they are growing rapidly, and they are making up a larger portion of their total load.

In recent years, with more and more large nonlinear power electronic converters being utilized, power system waveform distortion has warranted the development of stringent harmonic distortion control limits by several agencies around the world.

There are several reasons and incentives for reducing the harmonic current and voltage distortion produced by controlled rectifiers in the power system; perhaps the greatest incentives are:

1. utility and engineering specification imposed distortion limits (IEEE-519);
2. telephone interference complaints;
3. reducing power system equipment thermal and voltage stresses produced by the distortion.

11.3 Harmonics

Any steady-state periodic time domain waveform can be expressed by an infinite summation of sinusoidal waveforms at integer multiples of a fundamental frequency

The definition of power system harmonics is based on the application of the Fourier transform and superposition to voltage and current waveforms. An ideal power system contains only the first harmonic - 50Hz. When non-linear load conditions exist, the distortion of the voltage and current waveforms can be explained and analysed using the Fourier transform.

$$f(t) = \frac{a_0}{2} + \sum_{n=0}^{\infty} a_n \cdot \cos\left(n \cdot \pi \cdot \frac{t}{L}\right) + b_n \cdot \sin\left(n \cdot \pi \cdot \frac{t}{L}\right) \tag{11.3.1}$$

$$a_n = \frac{1}{L} \cdot \int_{-L}^{L} f(t) \cdot \cos\left(n \cdot \pi \cdot \frac{t}{L}\right) dt \tag{11.3.2a}$$

$$b_n = \frac{1}{L} \cdot \int_{-L}^{L} f(t) \cdot \sin\left(n \cdot \pi \cdot \frac{t}{L}\right) dt \tag{11.3.2b}$$

where f(t) = Time domain function;
n = Harmonic index;
L = Length of one cycle in seconds.

11.4 Sources of harmonics

Since harmonic distortion is caused by non-linear elements connected to the power system, anything that has non-linear characteristics will cause harmonic distortion. There are many kinds of harmonic producing loads available. Most loads have some degree of non-linearity, but in most cases their affect on the distribution system can safely be neglected. Some examples of common sources of power system harmonics some of which may not cause serious problems are [Arrillaga et al., 1985]:

1. Transformer saturation;
2. Transformer inrush;
3. Transformer neutral connections;
4. MMF distributions in ac rotating machines;
5. Electric arc furnaces;
6. Fluorescent lighting
7. Computer switch mode power supplies;
8. Battery chargers;
9. Imperfect ac sources;
10. Static VAR compensators;
11. VFD's;
12. DC converters;
13. Inverters;
14. Television power supplies.

One of the strongest sources of harmonic distortion are large VFD type loads. These loads not only have the potential to produce harmonic distortion, the amount varies with loading. It is because of these large harmonic producing loads and the affect they have on power systems that harmonic distortion limits have been developed.

11.5 Harmonic currents drawn by the VFD's

Information about the harmonic currents drawn by the VFD's is the key to any harmonic analysis. Characteristic harmonics are related to the pulse number by the following equation:

$$h = p \times n \pm 1 \tag{11.5.1}$$

where h = The harmonic order;
p = The pulse number;
n = An integer having values of 1, 2, 3, ...

Theoretically, this means for example that a six-pulse converter at the front end of a VFD draws harmonic currents of orders 5, 7, 11, 13, 17, 19, 23, 25, ... etc. Practically, however, noncharacteristic harmonics are also present due to slight irregularities in the conduction of the converter devices, unbalanced phase voltages, and other reasons. Therefore, a 12-pulse converter (theoretically only 11, 13, 23, 25, ... etc.) would also draw relatively small magnitudes of the 5^{th}, 7^{th}, 17^{th}, 19^{th}, ... etc., harmonics in addition to its characteristic harmonics. Because the VFD's are three-phase three-wire devices, the 3^{rd} and multiples of the 3^{rd} harmonic currents will not flow [Peeran et al., 1995].

Table 11.1 lists typical characteristic harmonic magnitudes for the most common pulse-number static power converters. However, inaccuracies in thyristor firing, differences in thyristor charcteristics, and system unbalances would cause the production of other *noncharacteristic* harmonic orders such as 3^{rd}, 4^{th} and 6^{th}.

Table 11.2: Harmonic currents present in input current to a typical static power converter in per-unit of the fundamental current Reprinted from [Shipp et al., 1996].

Converter	Order							
Pulses	**5**	**7**	**11**	**13**	**17**	**19**	**23**	**25**
6	0,0175	0,111	0,045	0,029	0,015	0,010	0,009	0,008
12	0,026	0,016	0,045	0,029	0,002	0,001	0,009	0,008
18	0,026	0,016	0,007	0,004	0,015	0,010	0,001	0,001
24	0,026	0,016	0,007	0,004	0,002	0,001	0,009	0,008

Six- and twelve-pulse converters are the most widely used. Three-pulse devices are used in small power applications such as extruders, and eighteen-pulse and above are used in much larger applications. The choise of pulse number is a matter of economics versus harmonic control.

Larger ac drives, 100hp and greater, typically use thyristor converters to control bus voltage or line current to maintain desired speed. The disadvantage of the thyristor converter is a greater ac current distortion during slower motor speeds. Fifth harmonic current distortions greater than 50% have been recorded at ac drive installations [Shipp et al., 1996]. The increase of harmonic current distortion is a function of the type of ac drive and its speed.

From the point of view of the harmonic currents drawn from the ac source, VFD's can be grouped into two basic types:
1. Type I has a front-end controlled rectifier using thyristors and GTO's; VCI and CSI drives belong to this type.
2. Type II has a front-end uncontrolled (diode rectifier); PWM drives belong to this type.

The harmonic content of the input currents of the two types VFD's depends primarily on the design of the VFD. The source impedance has but a second order effect on the harmonic content. Each VFD has what can be termed as a distinct harmonic *signature* which depends upon the type and the design [Peeran et al., 1995].

In order to characterize each VFD, the Harmonic Factor (HF) is introduced. This factor is defined as follows:

$$HF = \frac{\sqrt{25 \cdot I_5^2 + 49 \cdot I_7^2 + 121 \cdot I_{11}^2 + 169 \cdot I_{13}^2 + \ldots + h^2 \cdot I_h^2}}{I_1} \tag{11.5.2}$$

where HF = harmonic factor characterizing the VFD;
$I_5, I_7, I_{11}, \ldots, I_h$ = rms values of the 5^{th}, 7^{th}, 11^{th}, ...h^{th} harmonic currents;
I_1 = rms value of the fundamental (50Hz) current drawn by the VFD

In Table 11.2 guidelines are given for estimating the HF for VFD's whose performance is not known.

Table 11.2: Typical values of HF.

VFD Type and Size	HF
Large 12-pulse CSI and VCI VFD's 1000hp or more	0.5 - 1.0
6-pulse CSI and VSI VFD's up to 2500 hp	1.4 - 3.0
6-pulse PWM VFD's 5-250hp	2.0 - 5.0
6-pulse PWM VFD's 1-5hp	Approximately 12.0

The HF is related to the Total Harmonic Distortion (THD) of the voltage by the following equation:

$$\text{THD} = \text{HF} \times (\text{DrivekVA} / \text{SCkVA}) \times 100\% \qquad (11.5.3)$$

where DrivekVA = the full load kVA rating of the VFD
SCkVA = short-circuit kVA of the distribution system at the point of connection.

Figure 11.1 shows a plot of the THD against the ratio (DrivekVA / SCkVA) for various values of the HF. If 5% is the desired limit of the THD, then the hatched area in figure .1 shows the VFD ratings for which no harmonic filtering is necessary. If the system feeds many similar VFD's, then the aggregrate kVA rating should be used in the calculation of the percent THD. If many dissimilar VFD's are supplied from the same bus, then a knowledge of the current harmonic spectrum is required to calculate an equivalent HF.

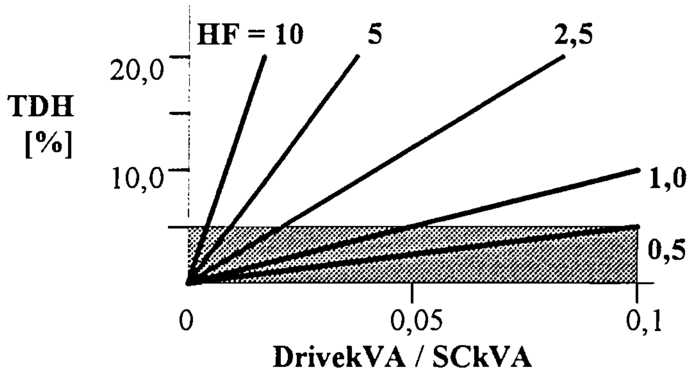


Figure 11.1: Percent voltage THD due to the VFD. Reprinted from [Peeran, 1995].

Table 11.3 lists the recommended harmonic voltage distortion levels for various voltage ranges. Even though the limits originally were for the utility point of common coupling, these values are used as a design guide in controlling harmonics within an industrial facility. If economics preclude phase multiplication techniques, larger sized harmonic filters can be installed to reduce harmonic magnitudes to acceptable levels.

Table 11.3: Harmonic voltage distortion limits (% of fundamental).

	< 69kV	69 - 161kV	> 161kV
Maximum for Individual Harmonic	3.0	1.5	1.0
Total Harmonic Distortion (THD)	5.0	2.5	1.5

11.6 Solving drive harmonic problems at the design stage

11.6.1 General

Harmonics associated with large VFD's or large groups of smaller drives can create severe problems for an industrial plant's power distribution system and the utility's power grid. The harmonics created by the drive fall into two categories:

1. The first one contains the harmonics that are related to the number of pulses of the drive and result from the rectifier current being approximately a square wave;
2. The second one contains high-frequency resonances that are due to the notching of the voltage waveform during the commutation period of the rectifier circuit.

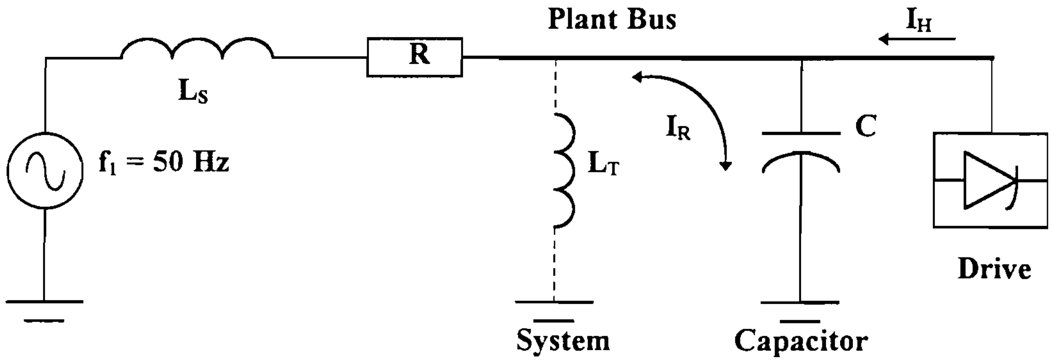
While harmonics can cause major equipment failures, unexplained trips, or communication noise, a drive system can be designed to avoid these adverse effects. This commonly means harmonic filtering is incorporated. A detailed specification for the drive and harmonic-reduction package must be created. The specification must be clear as to what the vendor is expected to provide in terms of equipment, design data, and calculations. The specification must also provide the vendor with accurate and complete data on the existing power distribution system. During the review of the vendor proposals, it is imperative to understand the methods being used to reduce the harmonics, so that technical inadequacies can be identified.

11.6.2 A first approach

All rectifier-based equipment such as drives or uninterruptible power supplies will create harmonic currents, which can invade a power system. There are several things a supplier can do to reduce the harmonics generated by a package, but they cannot be entirely eliminated.

A potential place for a failure due to harmonic heating is in the motor powered by the drive. Presently, many drives are being installed on standard induction motors. Here the problem of harmonic heating is compounded with reduced cooling by shaft driven fans, as the motor is operated below rated speed. This can be a major concern on constant-torque applications. Most large drives and sophisticated small drives utilize some combination of wave shaping and output filtering to reduce the harmonics impressed upon the motor. It is often best to provide the drive manufacturer with load speed-torque curves, motor data, and the speed range, and then give him the responsibility to determine if a potential heating problem exists. Additionally, temperature detectors must be used within the motor, as standard overload relaying is relatively insensitive to harmonics and cannot compensate for reduced air flow at lower speeds.

Once a piece of equipment such as a large drive is operated, any harmonic currents produced by the drive at the resonant frequency would result in harmonic currents flowing within the resonant circuit amplified many times larger than the drive exciting current. The effect of this is often observed as either capacitor can or fuse failures in another area of the facility or power system. The resonance is normally due to large capacitor banks resonating with the inductance of the transformers and motors. This situation can be eliminated by checking the power system during the early stages of design to make certain that no resonant circuits exist that are tuned near the frequency of harmonics produced by the drive. If any do exist, they can be detuned by either the addition of harmonic filters at the drive or alterations at the capacitor banks. Figure 11.2 shows a simplified first approach, where the drive harmonic currents I_h induce a resonance in the capacitor bank if they are at the same frequency f_h as the natural resonant frequency f_n of the capacitor/inductor network.



L_S = Source Inductance
L_T = Other plant transformer inductances

<u>Case 1:</u> **Ignoring other plant transformers**

Resonant frequency: $$f_R = \frac{1}{2 \cdot \pi} \sqrt{\frac{1}{C \cdot L_S}} \cong f_1 \cdot \sqrt{\frac{SystemS.C.MVA}{CapacitorMVAR}}$$

<u>Case 2:</u> **Including a plant transformer**

Resonant frequency: $$f_R = \frac{1}{2 \cdot \pi} \sqrt{\frac{1}{C \cdot \left(\frac{L_S \cdot L_T}{L_S + L_T} \right)}}$$

Figure 11.2: Calculated resonance in capacitor bank. Reprinted from [Dewinter et al., 1990]

11.6.3 Design check points

During the design process of the drive, various checks must be performed to ascertain that the drive system will perform as required without creating problems in the rest of the power system.

It is important to review with the vendor the harmonic reduction methods used and the calculated harmonic currents and voltages that will appear at the supply bus. The designer of the system should be able to explain adequately where the harmonic currents are circulating, based on the theoretical harmonic currents produced by the drive. Both current and voltage distortion should be reviewed on an individual harmonic and total distortion basis to verify that both have declining magnitudes with increasing harmonic order, indicating that no resonances are present. Also, if filters are present, the first tuned frequency should be at or below the first characteristic harmonic frequency. This will act to absorb that harmonic current while forcing the resonant frequency of the filter with adjacent loads to be below this characteristic frequency. The vendor should at this time be able to provide calculations giving the voltage notching which will be present on the supply bus based on the drive line reactance and the bus supply impedance. The series resonance frequency should have been calculated by the filter designer to verify that oversizing of capacitor banks will not be required to detune the power system/filter resonance away from integer harmonics. A reasonable design sizing can be achieved by having the voltage ratings on the capacitor cans higher than the arithmetic sum of the fundamental voltage plus the resonant voltage. The resonant voltage is the product of the resonant current times the capacitive reactance at the tuned frequency. The filter design should allow for detuning due to ambient temperature variations and supply frequency variations. The exact tuned frequency of each filter branch should not be at the frequency of a particular harmonic but rather slightly off to allow for some reactance in the circuit to a particular harmonic that will limit the current that will flow when resonance takes place [DeWinter et al., 1990].

At this time the power utility may need to be involved. In many locations the utility will insist on reviewing these calculations for the system. The utility can also provide useful feedback on the design as they often have in-house programs for modeling the power system harmonically. The power utility must also guarantee that their power system will not force more harmonic current into the filter than that allowed for by the filter designer (approximately 10%).

11.6.4. Field verification

Once the drive system has been installed, field testing should be done during commissioning to verify that the specifications have been met. Specialized equipment will be required for these tests, which can be performed with either plant staff and rental equipment or by a third party. Prior to energizing the filter, an analyser can be put on each branch to determine the impedance as a function of frequency and thus the tuning of the filter. Once the drive is operating, a brush or high-speed recorder should be used to plot the voltage and current at the drive input and the supply bus to determine notching and at each filter branch to determine current draw. A fast Fourier transform (FFT) or harmonic analyser shouls be used to record the voltage and current at the supply bus, each filter branch at the input of the drive. This will both verify the adequacy of the system design and prove invaluable for future troubleshooting comparisons.

11.6.5 Conclusion

Though the proper design and design checks, a large drive can be installed that will create no harmonic problems for an industrial power system. A well-designed system requires thet the purchaser provides the vendor not only a complete operational specification but also some key power system information. With care and attention to the design succes is certain.

11.7 Harmonic filters for VFD's

11.7.1 A design guideline

In the past, system designers simply relied upon the VFD manufacturer to assure them that there would be no harmonic problems. This was because the designers did not fully understand the harmonic performance of the system and the drives and because the VFD vendors usually excluded the information about the harmonic performance from their descriptive literature and application guides. Now that more and more users are becoming aware of the problem, it is in the best interest of the VFD manufacturers to help the designers assess the harmonic performance of the system. Presently, many architects and engineering consulting (A&E) firms require the VFD suppliers to perform a complete harmonic analysis, and determine the need for harmonic filters and size the harmonic filters, if required. Such a practice does not necessarily give the best design. This is because the A&E firm and the user have a better knowledge of the electrical distribution system, presently designed and possible future extensions, than the VFD manufacturer. Therefore, cooperation and exchange of information between the VFD manufaturer and the system designer will lead to a better system design.

11.7.2 Types of harmonic filters and tuning

Harmonic filters consist of several parallel sections of single-tuned of double-tuned series L-C resonant circuits. These are connected to the system preferably at the source side of the VFD isolation transformer or line reactors. It is well known that the series L-C circuit has the lowest impedance at its resonant frequency. Below the resonant frequency the circuit behaves as a capacitor and above the resonant frequency, as an inductor. Ideally, the filter must be tuned exactly to the characteristic harmonic it is supposed to suppress. Practically, however, the filters are tuned to a frequency slightly lower than the nominal resonant frequency in order to avoid the possibility of parallel resonance in case the filter component parameters change due to temperature and aging. Most low and medium voltage harmonic filters are usually tuned to about 0,95 times the nominal resonant frequency. Attenuation of the harmonic voltages produced by the filter reduces as the ratio of the actual resonant frequency to the nominal resonant frequency departs from 1,0.

The following basic equations relate to single tuned filters:

- Resonant frequency:

$$f_r = \frac{1}{2 \cdot \pi \cdot \sqrt{L \cdot C}} \tag{11.7.2.1}$$

- Detuning factor:

$$\alpha = \frac{f_r}{f_n} \tag{11.7.2.2}$$

- Impedance of the filter to the h^{th} harmonic current:

$$Z_f(h) = R_f + j\left[2 \cdot \pi \cdot f_1 \cdot h \cdot L - \frac{1}{2 \cdot \pi \cdot f_1 \cdot h \cdot C}\right] \tag{11.7.2.3}$$

- 60Hz kVAR of the filter capacitor:

$$\text{KVAF} = 0{,}377 \cdot C \cdot E_{L\text{-}L}^{\ 2} \tag{11.7.2.4}$$

where f_1 = Fundamental frequency = 50 Hz;
f_r = Resonant frequency of the filter, [Hz];
f_n = Nominal resonant frequency of the filter, [Hz];
h = Harmonic order;
R_f = Resistance of the filter circuits, [Ω]
L = Inductance of the filter coil, [H];
C = Capacitance of the filter capacitors, [F];
$E_{L\text{-}L}$ = Nominal line-to-line voltage of the system at the point of connection of the filter.

11.7.3 Attenuation of harmonic voltages by the filters

A harmonic filter attenuates all harmonic voltages at the point of its connection. Maximum attenuation occurs for the voltage whose frequency is equal to or close to the resonant frequency of the filter. The attenuation factor is defined as follows:

$$a_n(h) = \frac{V(h)}{V_f(h)} \tag{11.7.3.1}$$

where $a_n(h)$ = Attenuation factor due to the n^{th}-order filter of the h^{th}-order harmonic voltage;
V(h) = The h^{th} harmonic voltage without the filter at the point of connection;
$V_f(h)$ = The h^{th} harmonic voltage with the filter at the point of connection.

A high value of the attenuation factor $a_n(h)$ is desirable. A value equal to 1.0 means that there is no attenuation due to the filter. A value less than 1.0 means that there is amplification of the harmonic voltage instead of attenuation. A positive value of the attenuation factor at any frequency means that the impedance of the filter circuit is inductive at that harmonic frequency. A negative value means that the impedance is capacitive at that frequency.

Since α is generally less than 1.0, the resistance of the filter circuit has a minor effect on the attenuation produced by the filter. Therefore, neglecting the filter circuit resistance R_f as well as the resistance of the source impedance the following equation can be derived [Peeran et al., 1995]:

$$\alpha_n(h) = 1 + \left(\frac{(n \cdot \alpha)^2}{1 - \left(\frac{n \cdot \alpha}{h} \right)^2} \right) \cdot \left(\frac{kVAF}{SCkVA} \right) \tag{11.7.3.1}$$

The above equation gives the attenuation factor due to one single-tuned harmonic filter of nominal frequency of (n x f_1) for the h^{th} harmonic voltages at the point of connection.

If two or more filters of nominal frequency of n_1 x f_1, n_2 x f_2, ... etc., are connected in parallel, then the combined attenuation factor for the h^{th} harmonic voltage is given by:

$$a_F(h) = 1 + q_1(h) \times (kVAF_1 / SCkVA) + q_2(h) \times (kVAF_2 / SCkVA) \ldots \quad (11.7.3.2)$$

where

$$q_1(h) = \frac{(n_1 \cdot \alpha_1)^2}{1 - \left(\frac{n_1 \cdot \alpha_1}{h} \right)^2}$$

$$q_2(h) = \frac{(n_2 \cdot \alpha_2)^2}{1 - \left(\frac{n_2 \cdot \alpha_2}{h} \right)^2}$$

For example an eleventh harmonic filter alone produces an attenuation for the h^{th} (h = 5, 7, 11, 13, ...) harmonic voltage of

$$a_{11}(h) = \left(\frac{(11 \cdot \alpha_{11})^2}{1 - \left(\frac{11 \cdot \alpha_{11}}{h} \right)^2} \right) \cdot \left(\frac{kVAF_{11}}{SCkVA} \right) \tag{11.7.3.4}$$

Figure 11.3 shows the attenuation factors produced individually by this filter. This figure is drawn for α = 0,95 which is most usually the case. The value of *h* indicates the harmonic order of the voltage which is being attenuated.

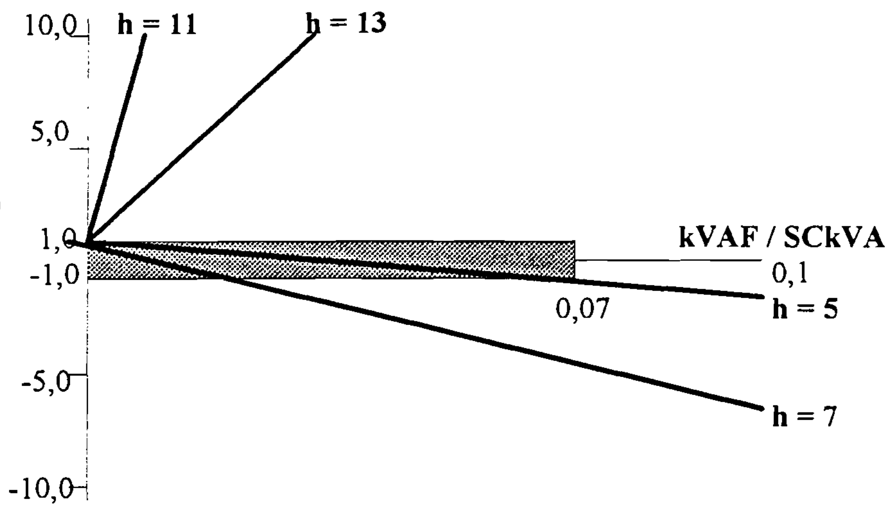


Figure 11.3: Attenuation factor due to an 11th harmonic filter. (α = 0,95, f_r = 627 Hz).

From figure 11.3 it can be concluded that an 11th harmonic filter when use alone appears as a capacitor to the 5th, and the 7th harmonic currents. For values of ($kVAF_{11}/SCkVA$) less than 0,07, the 11th harmonic filter actuals amplifies the 5th harmonic voltage instead of attenuating it.

11.7.4 Design of harmonic filters

Unfortunately, no simple generalized relation can be developed between the filter kVAR and the reduction in the voltage THD. Therefore the following procedure is suggested for the design of the filters using the equations given above [Peeran et al., 1995]:

1. Obtain the harmonic signature of the VFD from the manufacturer.
2. Calculate the voltage THD at the point of connection of the VFD and decide whether harmonic filtering is necessary.
3. If harmonic filtering is required calculate the percent harmonic voltages without the filter using the following equation:

$$(h) = \%I(h) \cdot h \cdot \left(\frac{DrivekVA}{SCkVA} \right) \tag{11.7.4.1}$$

4. Begin with a 5th harmonic filter. Select a 50Hz kVAR for the capacitor of approximately 1/3 or 1/4 of the VFD kVA rating. Calculate the attenuating factors $a_5(h)$ for h = 5, 7, 11, 13, 17, 19, ...
5. Calculate the harmonic voltages with the filter using the following equation:

$$_f(h) = \frac{V(h)}{a_5(h)} \tag{11.7.4.2}$$

6. Calculate percent THD with the filter. Some iterations are required at this stage to select the minimum rating of the filter such that the THD is just below the desired value (5% or less).
7. If the THD cannot be reduced below the desired value, add a seventh harmonic filter and repeat the design.

The design involves some trial and error steps based upon experience and judgement.

11.7.5 Specification of harmonic filters

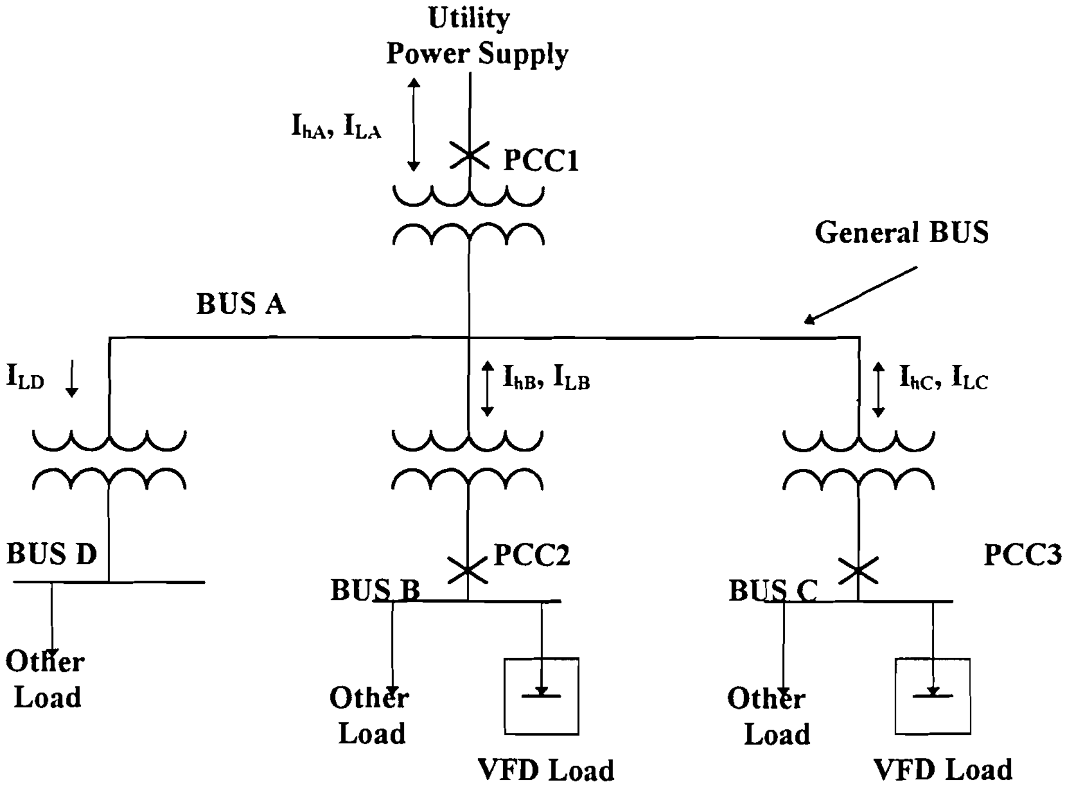
Specifying harmonic filters requires considerable care and attention to detail. Several additional calculations are required before one is ready to write the specifications of the filter. Broad guidelines for writing the specifications for the filters are as follows:

1. It is best to specify one filter for each VFD for VFD's rated 25hp or more.
2. Each filter must have a contractor at the input which is controlled by the VFD such that the VFD is connected only when the VFD is running. In order to avoid any switching transients while the motor is starting, the filter contractor must be closed after a certain time delay after the motor reaches the set speed. This will also provide time for discharging the filter capacitors before the filter is reconnected.
3. The filter capacitors will be subjected to a voltage more than the system voltage because of the flow of the harmonic currents in addition to the fundamental frequency current. Therefore the voltage rating of the capacitors must be more then the distribution system voltage.
4. Filters are most effective when connected at the primary of the VFD isolation transformer. A fused disconnect or a circuit breaker is required to provide short-circuit protection.
5. It is best to use an isolation transformer or line reactors between the VFD and the distribution system even for small VFD's. This is an inexpensive way of solving many problems. It reduces the voltage notches, eliminates high-frequency ringing and limits the available short-circuit current at the VFD.
6. Filter inductors can be air-cored or iron-cored. Iron-cored inductors are preferred because of their smaller size. The core must include sufficient air gap to prevent magnetic saturation at the maximum expected current.
7. Filter capacitors must be individually fused and monitored by overvoltage detecting circuits. The filter contractor must be tripped by these circuits.
8. The filter and the VFD specifications must require the supplier to perform harmonic measurements during shop tests and fields tests.

11.8 Methodology for computation of distortion to meet IEEE standard 519-1992

11.8.1 General

Figure 11.4 shows a typical industrial power system that includes large VFD loads at various locations, namely busses *B* and *C*. The standard defines such busses as dedicated busses. Bus *D* has no VFD loads. Bus *A* is defined as a general bus. The main point of common coupling (PCC) where current and voltage distortion is to be monitored is PCC1, located at the customer utility interface, usually the primary side of the main power transformer. For monitoring distortion, several other PCC's can be considered as shown in figure 11.4 at locations PCC2 and PCC3.



Note: Subscript h indicates harmonic # (5,7,11,13, ...)
Subscript LA denotes load current at BUS A.

Figure 11.4: Typical industrial power system with VFD's.

The methodology for computation of distortion can be summarized in the following steps:

Step 1: Individual current harmonic distortion at each dedicated bus can be computed by using manufacturer supplied computer programs or table 11.1 (approximate values).

Step 2: Once the harmonic currents are established at each dedicated bus, total harmonic currents at PCC1 for each individual harmonic can be computed as follows:

$$I_{hA} = I_{hB} + I_{hC} \tag{11.8.1}$$

where h = 5, 7, 11, 13, ...

In other words, each individual harmonic current at PCC1 is the sum of harmonic current contribution from each dedicated bus. If the currents are not in per unit, care should be taken to convert amps to proper voltage base. Similarly, the load current at PCC1 is given by:

$$I_{LA} = I_{LB} + I_{LC} + I_{LD} \tag{11.8.2}$$

The best way to estimate the maximum demand load current at PCC1 is to compute connected load for each branch feeder and multiply by a demand factor to obtain feeder demand. Then divide the sum of all feeder demands by a diversity factor to obtain the maximum demand load current [Keskar, 1996].

Step 3: Choose a base MVA and base kV for the system. Use the following equations [IEEE standard 519,1992] to compute individual and total current and voltage harmonic distortions at PCC1 and other desired points

$$I_b = \frac{MVA_b \cdot 10^3}{\sqrt{3} \cdot kV_b} \quad \text{[A]} \tag{11.8.3}$$

$$Z_S = \frac{MVA_b}{MVA_{SC}} \quad \text{p.u.} \tag{11.8.4}$$

$$_H = \frac{I_h}{I_b}(h) \cdot (Z_S) \cdot 100\% \quad \text{[V]} \tag{11.8.5}$$

$$THD = \left(\sum_H V_H^2\right)^{\frac{1}{2}} \% \quad \text{[V]} \tag{11.8.6}$$

$$I_H = \frac{I_h}{I_L} \cdot 100 \quad \text{for h, H} = 5, 7, 11, 13, .. \tag{11.8.7}$$

$$ITHD = \left[\frac{\sum_h I_h^2}{I_L^2}\right]^{\frac{1}{2}} 100\% \quad \text{where h} = 5, 7, 11, ... \tag{11.8.8}$$

$$S.C.ratio = \frac{I_{SC}}{I_L} = \frac{MVA_{SC}}{MVA_D} \tag{11.8.9}$$

$$k - factor = \sum_h (h)^2 \cdot \left(\frac{I_h}{I_L}\right)^2 \text{ where h} = 1, 3, 5, 7, \ldots \tag{11.8.10}$$

Variables in the above equations are defined as follows:

- I_b = Base current [A];
- kV_b = Base kV;
- MVA_b = Base MVA;
- MVA_{SC} = short circuit MVA at the point under consideration;
- Z_S = system impedance in P.U.;
- V_H = percent individual harmonic voltage distortion (h, H denote harmonic number);
- I_h = individual harmonic currents [A];
- I_H = percent individual harmonic current distortion;
- VTHD = Voltage Total Harmonic Distortion;
- I_{SC} = short circuit current at the point under consideration;
- I_L = estimated maximum demand load current;
- ITHD = Current Total Harmonic Distortion;
- MVA_D= demand MVA.

Values of individual and total voltage and current harmonic distortion can be computed with the above equations and compared with the IEEE limits. Equation (11.8.10) can be used to calculate the K-factor, which is useful in transformer design. The K-factor should be specified for transformers that feed VFD loads.

Step 4: If the analysis is being performed for CSI-type drives, area of voltage notch A_N should also be computed. Figure 11.5 shows an impedance diagram of the industrial power system shown in figure 11.4.

The procedure outlined here for notch area computation is very conservative and will lead to larger values of notch area: Multiple drives at each dedicated bus are lumped into one single large drive. The notch area at PCC1 (primary side of main transformer) is assumed to be the algebraic sum of notch area contribution from each dedicated bus. This assumes that all SCR's are synchronized and commutate such that the effect of notch width is additive.

Notch area A_{NDR1} at the drive connected to bus B can be computed as follows [Schieman et al., 1985]:

$$A_{NDR1} = 2 \cdot (L_S + L_{MT} + L_{F1} + L_{DT1} + L_{DR1}) \cdot I_{DR1} / 0{,}85 \text{ V}\mu\text{sec.} \tag{11.8.11}$$

where I_{DR1} is the combined full load input current of all drives connected to bus *B*. The inductances are in microhenries/phase referred to 480V, and the current is in amperes. Notch area A_{N1} due to bus *B* at PCC1 is given by:

$$A_{N1} = \frac{L_S}{L_S + L_{MT} + L_{F1} + L_{DT1} + L_{DR1}} \cdot A_{NDR1} \tag{11.8.12}$$

Similarly, A_{N2} from bus *C* is calculated. The total notch area at PCC1 is estimated as follows:

$$A_N = A_{N1} + A_{N2} \tag{11.8.13}$$

Notch area at dedicated bus *B* is computed by the following equation:

$$A_{ND} = \frac{L_S + L_{MT} + L_{F1} + L_{DT1}}{L_S + L_{MT} + L_{F1} + L_{DT1} + L_{DR1}} \cdot A_{NDR1} \tag{11.8.4}$$

The notch area computation procedure for bus *C* is similar to the procedure outlined above. In Table 10.1 for a special application: A_N should be less than 16400 Vμsec.

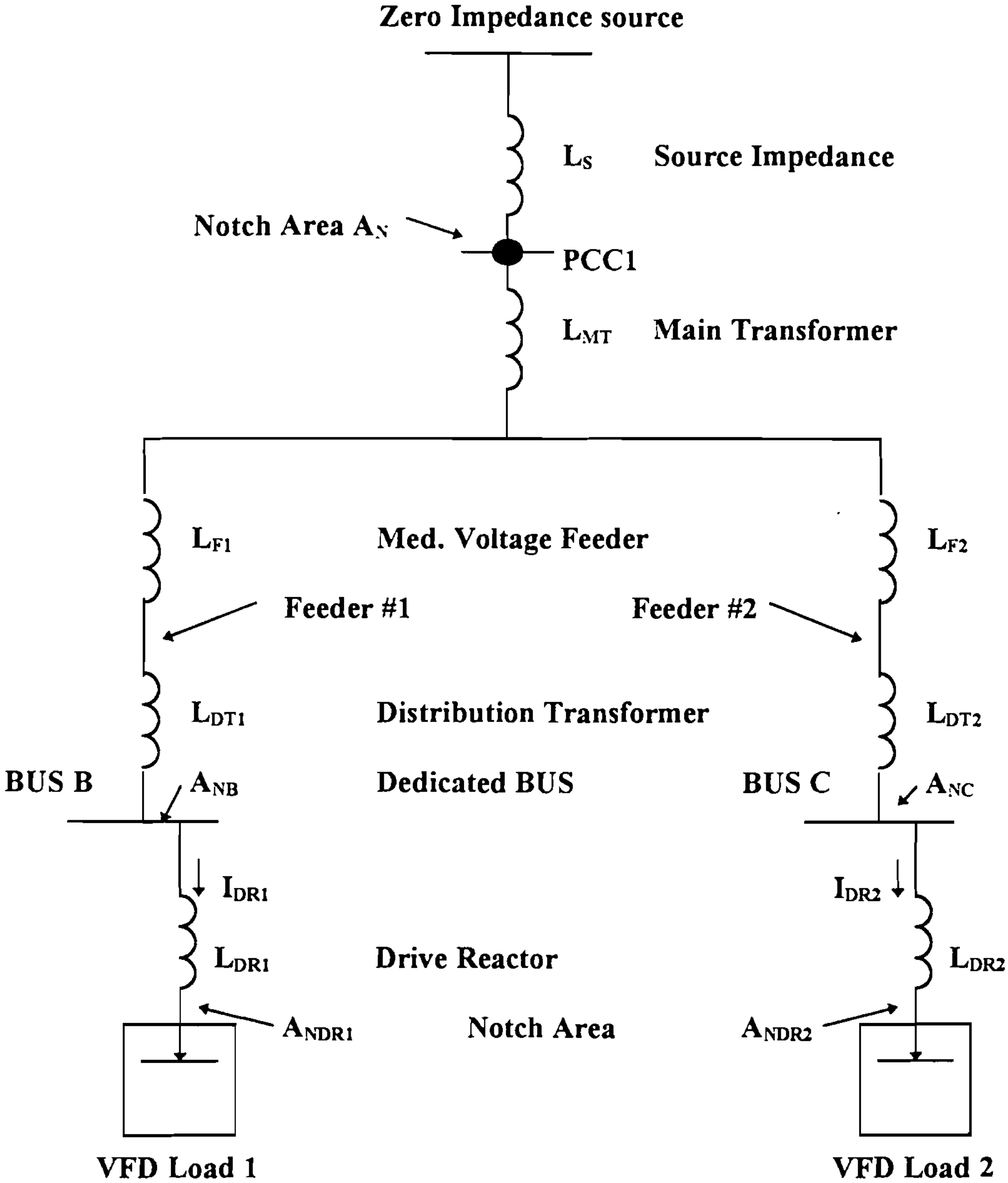


Figure 11.5: Impedance diagram of the industrial system shown in figure 11.4. (Feeders with no VFD load are omitted).

11.9 Current trend in VFD market

As a result of the new limits on the current harmonics, the VFD market is moving toward higher pulse converters as front ends. Twelve- and 18-pulse converters are readily available. Twenty-four- and 30-pulse drives are under development. Some manufacturers are offering *clean power controllers* that boast zero harmonic generation.

11.10 Conclusion

The new IEEE Standard 519 requires careful harmonic analysis of the entire power distribution system for specifying proper VFD technology. It is not sufficient to look at one location: the combined effect of all VFD drive locations on the PCC must be considered. It is recommended that the primary side of the main plant transformer should be considered as the main PCC where individual and total current and voltage harmonic distortion should be evaluated. As another check, VTHD should be evaluated at each dedicated bus where VFD's are connected. Each dedicated bus should be isolated from the rest of the system by a transformer. The transformer manufacturer should be informed of the K-factor for the current harmonics. Based on this analysis, proper VFD technology should be specified. The specification should require detailed calculations of the distortions. A plant single-line diagram (the single-line diagrams of AFS are given in Appendix F) with all necessary values of impedances, short circuit current, related ratings, and the location of distortion computation should be included in this specification. Last, but not least, a comprehensive performance test should be specified that requires measurement of distortion at specified locations in a plant running at specified load.

IEEE Standard 519 deals with the quality of input power supply. There are several other factors (for example: power quality in the motor circuit, efficiency, and effect of drives on the motor) that should be considered while specifying VFD's.

Chapter 12

Energy usage evaluation

12.1 General

It is probably save to say that in the majority of existing constant-speed drive applications in industry, *some* energy can be saved with the use of VFD's. Unfortunately, determining the amount of energy saved is normally a complex analysis with a result that can vary dramatically between applications [Rice, 1988]. Because ac drives are a sophisticated and costly piece of equipment, the degree of energy savings in a given application must be sufficiently large to justify a relatively large initial expenditure. This will reduce the number of applications where ac drive retrofits will yield an adequate economic payback.

Consequently, it is critically important that a *realistic* energy savings evaluation should be conducted for every VFD application under consideration to insure that an acceptable rate of return exists on the required capital expenditure. In order to do an energy evaluation, the electrical engineer must have an appreciation for the process and mechanical aspects of the drive installation.

There are several basic issues associated with conducting an VFD energy-savings evaluation, which may be summarized as follows:

1. A list of the information that is required to conduct the study should be compiled;
2. A source for all the required information should be determined. The procedures and assumptions necessary to do this should be established, recognizing that some data may have to be estimated;
3. The calculation procedures used to arrive at the energy savings must be determined. The method by which this power savings is compared against the initial capital investment must be established to determine the economic viability of a project;
4. A number of cases may have to be examined in order to consider various operating conditions, or to look at best/worst case analysis because of the wide tolerances in the input data;
5. All calculations and conclusions need to be summarized in a concise yet complete and rendable fashion.

Centrifugal pumps have enjoyed widespread use in industry in conjunction with constant-speed electric motors as drivers. Flow control has traditionally been accomplished in the process by methods involving a considerable expenditure of energy as a necessary trade-off for having that control. VFD's provide the opportunity of allowing the same (or greater) degree of flow control without the losses in the process.

Unfortunately, there are some real pitfalls in some analysis techniques which most often, will yield overly optimistic results. In order to avoid these pitfalls, energy savings should be determined by separately calculating the total energy usage from the constant-speed motor system and the VFD system. The energy savings is then the difference between the two energy usage calculations. Only in this way can all the differences in the application be evaluated [Rice, 1988]. This method will be applied in the calculations at the AFS system.

12.2 Pitfalls

12.2.1 Reason for concern

Using only (3.2.1.3) for calculating energy savings without modeling the centrifugal pump performance curve and/or without considering the system charcteristics will result in projected power being related to the cube of the speed reduction only. This symplistic view of the reduced speed performance does not consider the characteristics of the system, changes in the fluid density nor the efficiency characteristics of the centrifugal pump throughout its full performance range.These three factors can be significant and ignoring them can lead to overstated energy savings by as much as 50 percent or even more depending on the relative influence each factor has on the evaluation of power savings [Kempers, 1995].

Using the relation mentioned above for system evaluations can only be fully correct if:

- the fluid density doesn't change;
- the pump efficiency remains constant;
- the system characteristics result in the relationship 3.2.1.1

This means the temperature must remain constant throughout the flow range and the system must be made up of frictional losses only. No static head can be present in the system characteristic! Therefore system applications that have a fluid elevation or backpressure to consider will experience accuracy problems by using the relation 3.2.1.3.

12.2.2 Factors affecting errors

The higher the fluid pressure (referred to technically as static head offset) the larger the error. Unfortunately there is another factor to consider that affects the amount of error that can be realized. This factor is related to the efficiency curve for the pump throughout the flow range of the pump. The higher backpressure causes a thorough evaluation to cross through several efficiency points on the full speed efficiency curve. This complication means the evaluation must consider all of these factors to achieve reasonable accuracy in reduced speed performance predictions related to power and energy savings [Kempers, 1995].

12.2.3 The magnitude of errors

During the evaluation process to characterize the magnitude of errors related to introductions of static head offsets, it was discovered that yet another factor affects this inaccuracy. This additional factor is related to the relative location of the design flow to the best efficiency flow of the pump. Generally speaking, the errors are greater if the design flow is to the left of the best efficiency flow. Starting with a design flow greater than the Best Efficiency Flow (BEF) causes the evaluation to gain efficiency as the static head is increased, thus mitigating the net of errors.

With so many factors affecting the accuracy of the results, it is not possible to use general rules-of-thumb to correct for these conditions. Table 12.1 shows the magnitude of errors that can be introduced as the static head is changed from 0 percent of design head to 100 percent of design head.

Table 12.1: Errors due to Static head offset.

Static head offset in % of design head	**Percent Error for Design 'A'-flow = 56% BEF**	**Percent Error for Design 'B'-flow = 100% BEF**	**Percent Error for Design 'C'-flow = 144% BEF**
100	90	72	43
75	65	53	31
50	40	32	20
25	20	15	10
0	0	0	0

12.2.4 System head characteristics

Figure 12.1 shows the head vs flow system curves for each of five static head offset case studies and the full speed pump head curve as well. This example is for the condition where the design point flow is 144 percent of the BEF.

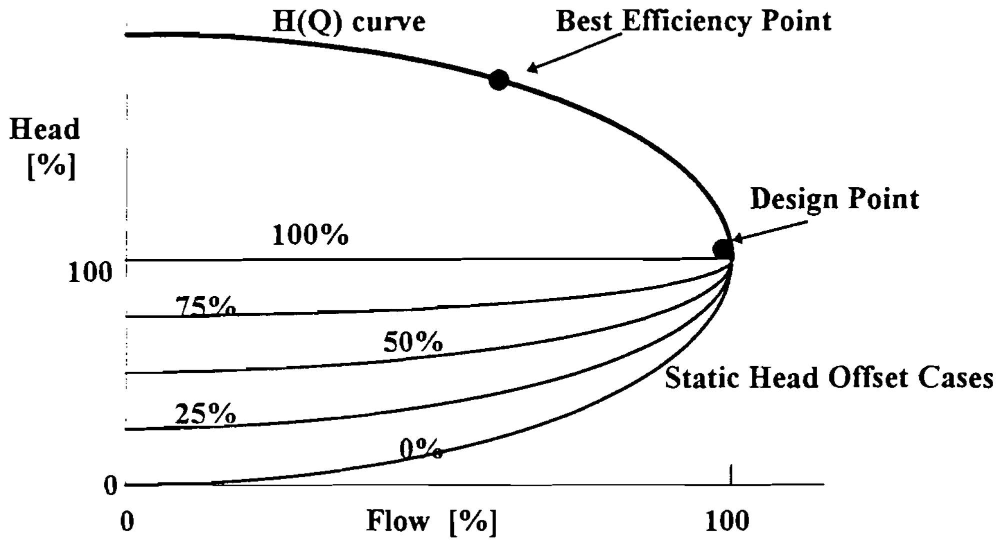


Figure 12.1: System head curves with static head offset cases
Shown for design condition 'C' only.

12.2.5 Power curves, savings, and errors for each case

Figure 12.2 shows the power requirements for each of the five static head offset cases and also for the full speed throttling condition. The vertical distance from the top curve (throttling) to each of the lower curves (cases) represents a power savings because of reduced speed operation. The best power curve is the 0 percent static head system curve because it shows a cubic relationship to speed reductions.

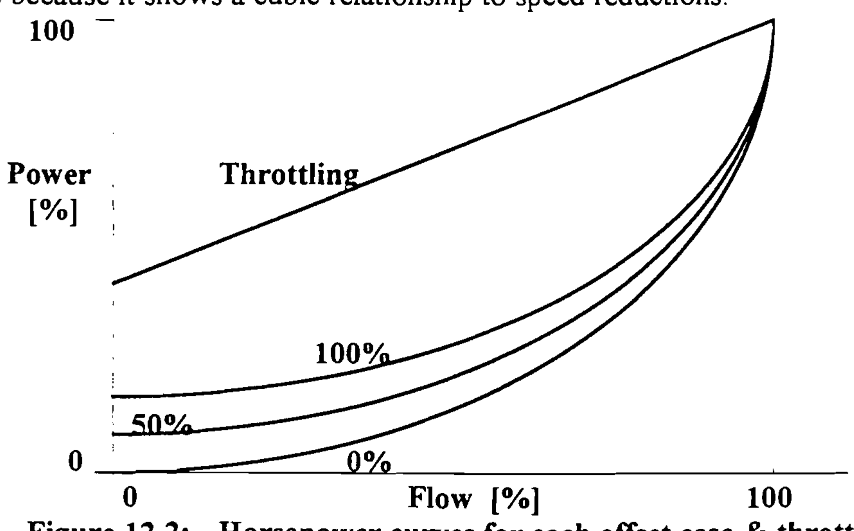


Figure 12.2: Horsepower curves for each offset case & throttling.
Shown for design condition 'C' only.

Figure 12.3 shows the savings that can be obtained for each case study. The amount of error from assuming that equation 3.2.1.3 would then be the vertical distance from the top curve (0 percent offset) and each of the other curves. The higher the static head the larger the distance from the top curve.

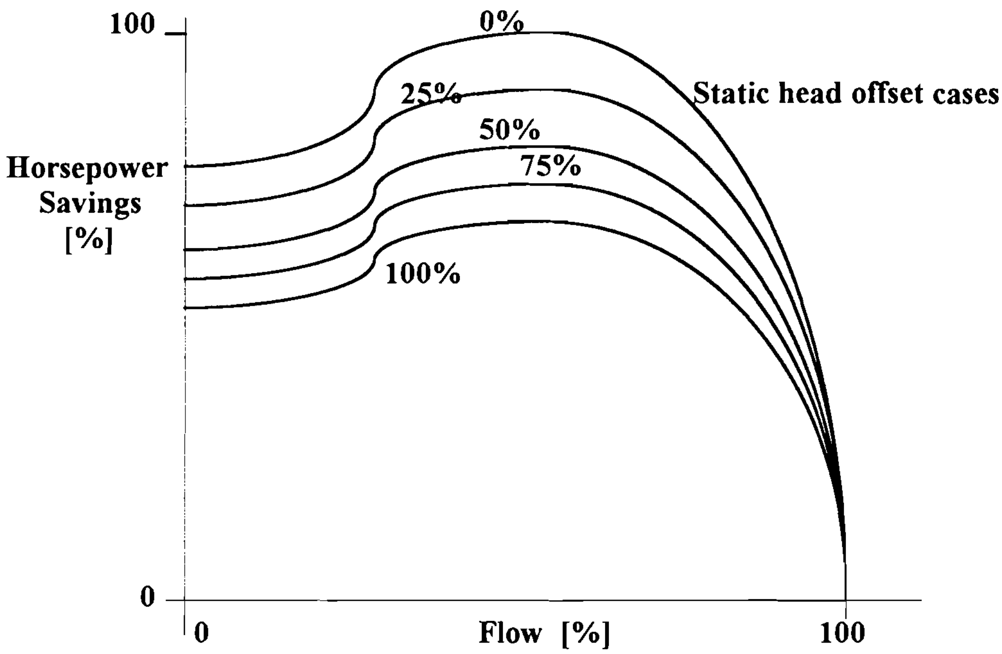


Figure 12.3: Power savings for each case compared to throttling. ***Shown for design condition 'C' only.***

It should be noted that even at the maximum static head offset of 100 percent full design there are still considerable savings in power and therefore energy over time. It should be clear from this figure that the savings are not the same for each flow point. So the power savings are sensitive to system flow conditions and calculations of energy savings must consider the amount of time that is spent at each of the flow conditions usually over a full year (Duty Cycle).

Figure 12.4 shows the resulting errors that can be found by ignoring the static head offset ('C' design flow). A brief summary of all cases A,B and C can be found in table 12.1. It is noticed that if the static head offset is larger, the error is larger.

12.2.6 Pump efficiency characteristics

Figure 12.5 shows the effect on the efficiency as result of each case for the 'C' design flow. This set of curves is different from the 56 percent and 100 percent of BEF design conditions because the efficiencies for the 'C' design go higher than the design point efficiency. The evaluation of the higher static head offset cases pass through and intersect the full speed efficiency curve as the speed is reduced. This gain in pump efficiency is what causes the errors to be smaller for the 'C' design condition [Kempers, 1995].

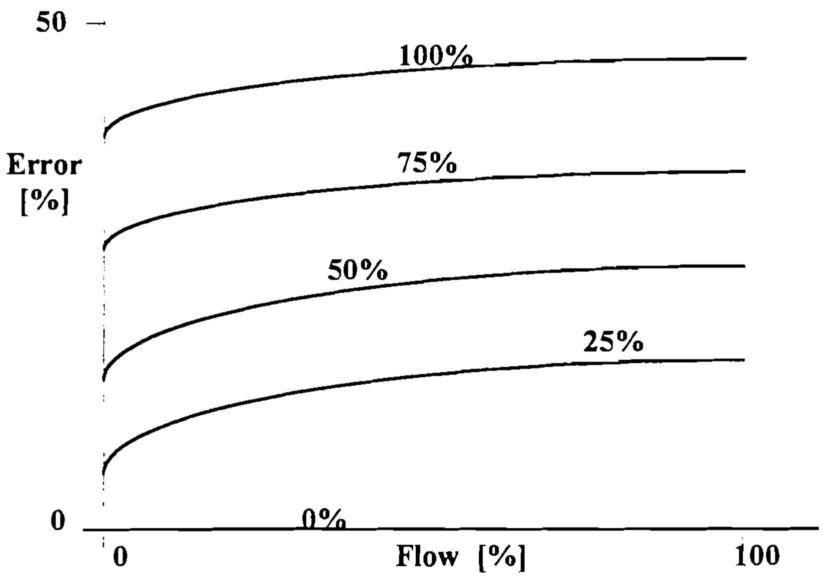


Figure 12.4: Percent error in savings projections.
Shown for design condition 'C' only.
reprinted from [kempers, 1995]

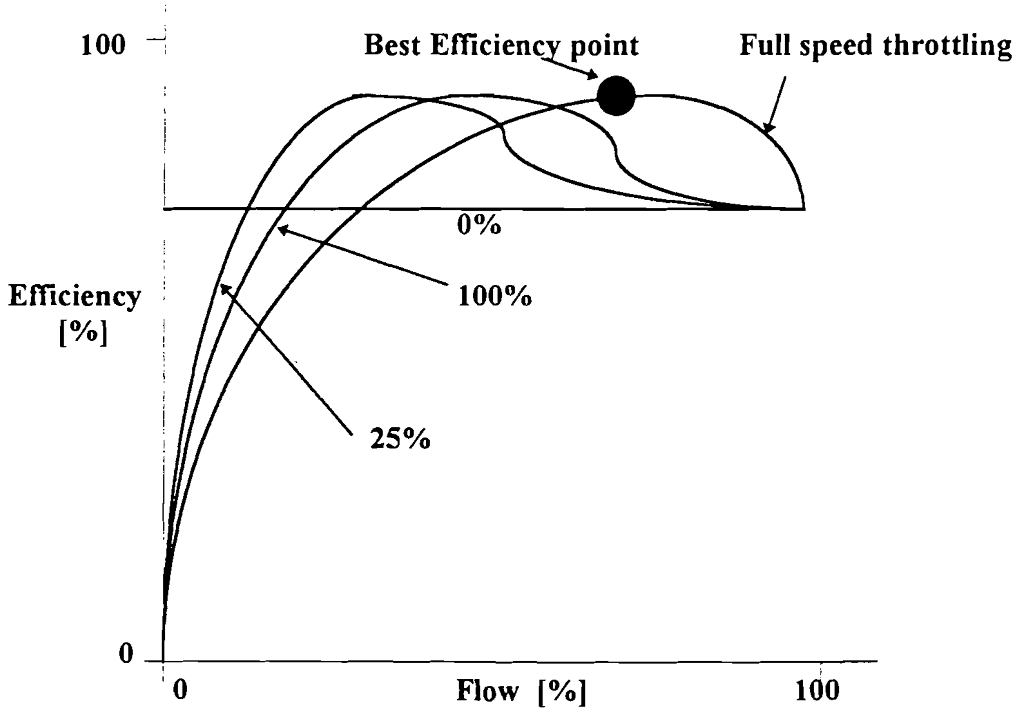


Figure 12.5: Pump efficiency curves for each case and Throttling.
Shown for design condition 'C' only.
reprinted from [kempers, 1995]

The other two ('A' and 'B') design conditions each start at the design flow efficiency and either stay constant as in the 0 percent case or fall off as the flow and speed are reduced. Thus these have larger errors (see table 12.1). It should also be noted that the 'B' design condition starts out with the best efficiency to begin with. The 'A' design condition starts out with an efficiency similar to the 'C' design condition but falls off in efficiency instead of gaining efficiency.

12.2.7 Speed for each case

Figure 12.6 shows the speed for each case that results from reduced speed operation in relation to the various static head offset cases. It is noticed that:

- the speed curve for the pure frictional case (0 percent offset) is a straight line from the design flow point to zero flow. This follows the relationship 2.3.1.2;
- the full speed throttling case at full speed is a straight horizontal line to the left of the design flow point.

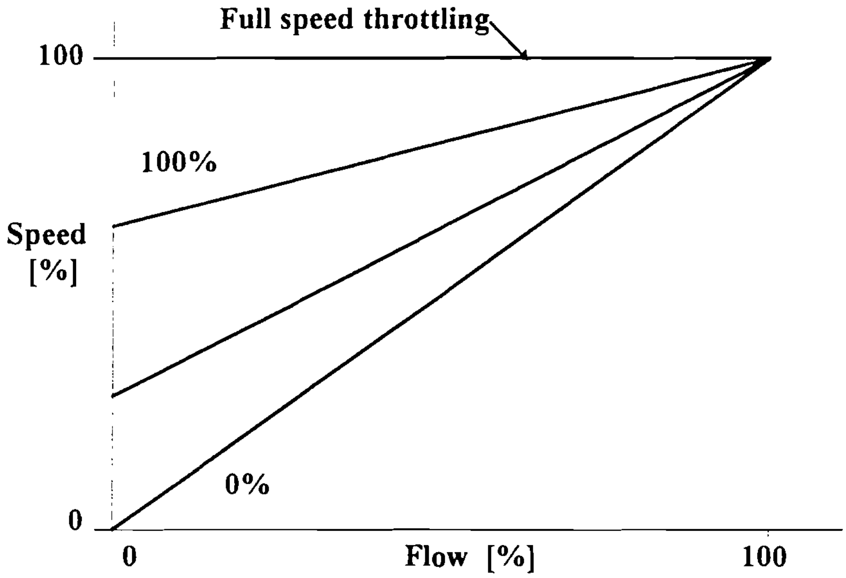


Figure 12.6: Pump speed for each case and Throttling.
Shown for design condition 'C' only.

The rest of the cases show speed curves that are between these two extreme cases. Higher static head offsets will result in higher speed curves. The relationship of speed to flow reduction is not linear for these cases.

12.2.7 Conclusions

Larger applications of VFD's require more expensive VFD's, and payback expectations become more important. By considering the savings one should be aware of potential pitfalls that can cause evaluation errors.

The use of equation 2.3.1.3 as the only evaluation tool for centrifugal pump reduced speed performance determination will limit the scope of accurate evaluations

The manufacturer's original centrifugal pump performance curves are needed to begin an evaluation. The evaluation can only continue if the performance data is modelled accurately and the system characteristics are understood and decomposed into the static head and frictional head components.

The need for accuracy in any study of energy savings due to reduced speed operation made available by the use of VFD's is related to the type of the system and the size of the application. The size will drive the cost of the VFD and the potential savings while the type of system will dictate whether any of these pitfalls may be present for consideration. These applications should also then consider the efficiencies of the VFD and motor while running at reduced speed and power levels. Corrections for deviations from the equations should be considered if possible.

12.3 Required data

The following is a summary of the information that is required to evaluate the potential energy savings for the use of VFD's in centrifugal pump applications:

1. *Method of flow control to which adjustable speed is to be compared:*
   a) output throttling **(AFS)**
   b) recirculation
   c) adjustable speed coupling
2. *Pump or fan data:*
   a) Head-versus-flow curve for every different type of liquid that is handled; in many applications only one product is handled and only one curve is required (at AFS this is only kerosine).
   b) Pump efficiency curves (often on same graph with the head/flow curve).
3. *Process information:*
   a) Specific gravity of the product ($\rho_{kerosine} = 840$ kg/m^3);
   b) System resistance head/flow line or precent static head and rated head and flow requirements;
   c) Equipment duty cycle, i.e., flow levels and time duration.
4. *Efficiency information on all relevant electrical system apparatus in sufficient detail to allow part load-loss analysis, including:*
   a) motors, constant and adjustable speed;
   b) ac adjustable-speed drives;
   c) gears;
   d) transformers.

It is not always possible to obtain all of the preceding information. Although precise information is preferable, reasonable assumptions can normally be made for the data in the preceding parts *(2)* and *(4)*. However, if the information in parts *(1)* and *(3)* is not known, there is simply no way to conduct a valid energy usage evaluation. Variances from the actual for these data variables will have a major affect on the magnitude of the calculated energy savings.

12.4 Energy evaluation procedure

The important and difficult part of an adjustable-speed energy-savings analysis is the data gathering and determination of the shaft horsepower required by the pump under the various operating conditions. The following is a summary of a suggested evaluation procedure from that point on.

1. Calculate the required horsepower for each flow case for adjustable speed and the base comparison system.
2. For each flow condition for the adjustable speed and the base comparison system, determine the actual load efficiencies of the electrical components of each drive system and calculate the total input.
3. Determine the total kilowatthour (kWh) usage for each flow condition for adjustable speed and the base comparison system.
4. Determine the total annual energy consumed by adding up the kWh at every flow condition for the adjustable speed and base comparison case. The difference between the total energy consumption for each case is the energy savings.

12.5 Energy evaluation of RijkI

In this paragraph the energy savings for RijkI, when VFD's are implemented, will be calculated. First the energy usage at this moment (the base comparison system) will be calculated. This will be done by the energy evaluation method mentioned above.

The following conditions are satisfied:

- Only fixed speed pumps;
- A new pump is started when the demanded flowrate $\geq$ m $\cdot$ 288 m^3/h, where m is the number of pumps already in operation;
- $Q_{totaal} = Q_{FSP} \cdot m$, where Q_{FSP} is the flowrate in one fixed speed pump;
- Calculations are performed with (3.5.2.4) - (3.5.2.6):
  $H = 148{,}8 + 0{,}0142 \cdot Q - 3{,}71 \cdot 10^{-4} \cdot Q^2$
  $P = 0{,}2837 \cdot Q - 1{,}75 \cdot 10^{-4} \cdot Q^2 + 39$
  $\eta = 2{,}29 \cdot 10^{-3} \cdot Q \cdot \dfrac{H}{P}$
- As switching to a new pump happens at (m $\cdot$ 288 m^3/h), the minimum produced head in the system is always H(288) = 122,1 m kersosene, what means a pressure of 9,9 bar (1 bar = 13,6/0,84 * 0,76 m = 12,3 m kerosene).
- Total efficiency (except the pump) is estimated at 0,95.

- Energy usage = E_{totaal} = E(50) + E(75)... + E(2150) =
  = $(P(50) / \eta) \cdot T(50) + (P(75) / \eta) \cdot T(75) + \ldots + (P(2150) / \eta) \cdot T(2150)$

In Appendix G1 the energy usage satisfying the conditions given above, is calculated. The total energy usage in one week at this moment at RijkI is approximately 29400 kWh a week.

The duty cycle of RijkI, recorded in February 1997 is given in figure 12.8.

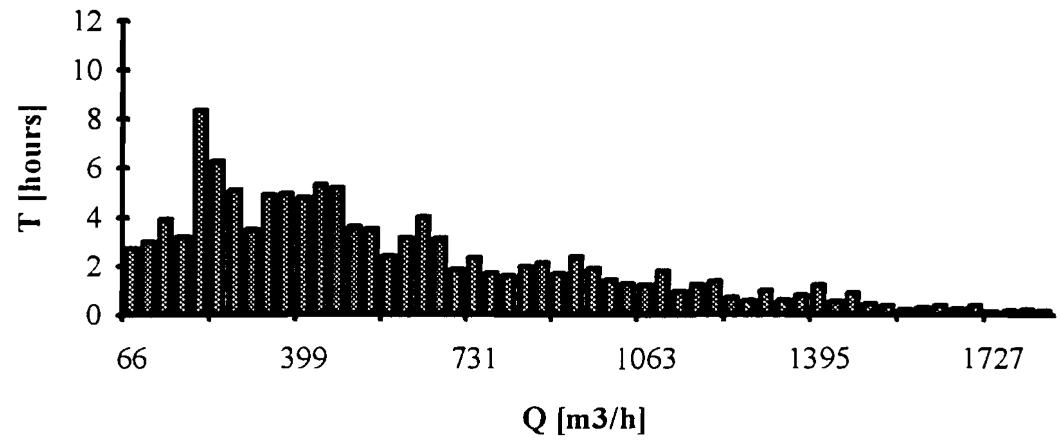


Figure 12.8: The duty cycle of the pumps at RijkI [recorded February 1997].

In may 1997 a new duty cycle was recorded. This is given in figure 12.9.

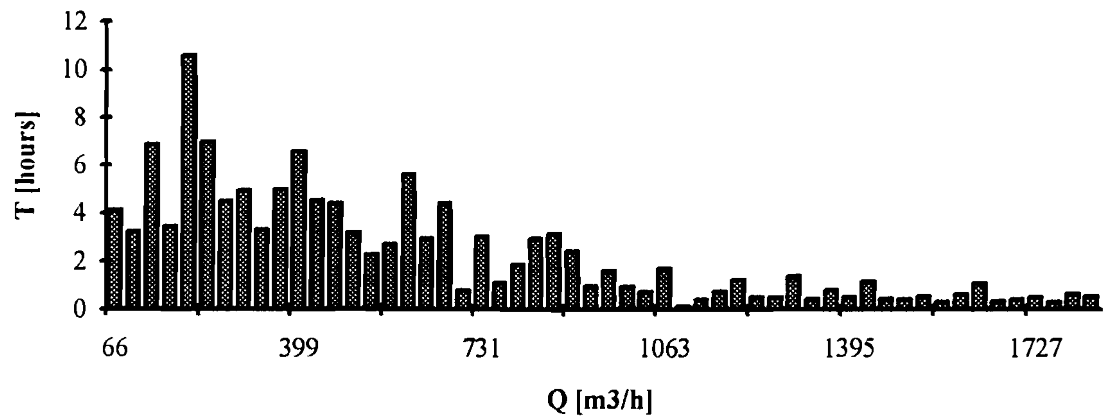


Figure 12.9: The duty cycle of the pumps at RijkI [recorded May 1997].

It is obvious from figure 12.8 and 12.9 that the duty cycle remains the same, only the peaks will be bigger (larger throughput), so it can be used for a realistic energy evaluation.

Now the VFD's will be introduced so that energy savings are possible. Eight different situations will be discussed (1 VFD, 2 VFD's, ... , 8 VFD's). The following conditions have to be satisfied:

- The minimum pressure at the hydrants is 8,5 bar = 104,6 meter kerosene;
- Number of VFD's: l = number of Variable Speed Pumps (VSP's);
- Number of FSP's: m;
- $Q_{totaal} = Q_{V1} + ... + Q_{Vl} + Q_{F1} + ... + Q_{Fm}$
  where $Q_{V1..l}$ = flowrate in a VSP;
  $Q_{F1..m}$ = flowrate in a FSP;
- Total efficiency (motor, VFD, coupling etc.) = 0,95.
- Q_{FSP} as high as possible and constant: 288 m^3/h.
- $Q_{V1} = Q_{Vl}$, $H_{FSP} = H_{VSP}$

So the following formulas are the most important:

- $Q_{totaal} = Q_V \cdot l + 288 \cdot m$
- H_{VSP} = 122,1 meter
- $H_{VSP} = (171 \cdot n^2 + 48{,}1 \cdot n \cdot Q - 3710 \cdot Q^2) \cdot 10^{-7}$ (3.5.2.1)
- $P_{VSP} = (326 \cdot n^2 \cdot Q - 593{,}3 \cdot n \cdot Q^2 + 15{,}2 \cdot n^3) \cdot 10^{-10}$ (3.5.2.2)
- $\eta = 2{,}29 \cdot 10^{-3} \cdot Q \cdot \dfrac{H_{VSP}}{P_{VSP}}$ (3.5.2.3)
- Energy saving = $E_{saving} = E_{oud}(Q) - E_{nieuw}(Q)$

For the situation where only VSP's are in operation, the necessary pressure at the hydrants is equal to:

$$H_{necessary} = 104{,}6 + H_{vtot}(Q) - H_{tank} \qquad (12.5.1)$$

where H_{vtot} = Total losses in the system (pipes, valves) [m]
H_{tank} = Height of the tank where the fuel is pumped out.

The values for the different heights of the tanks is:

Rijk1B:	*Rijk2B:*	*Combination:*
$H_{tankmax}$ = 14,6 meter	$H_{tankmax}$ = 16,9 meter	$H_{tankmax}$ = 15,8 meter
$H_{tankhalf}$ = 7,8 meter	$H_{tankhalf}$ = 9,1 meter	$H_{tankhalf}$ = 8,5 meter
$H_{tankmin}$ = 1,0 meter	$H_{tankmin}$ = 1,3 meter	$H_{tankmin}$ = 1,2 meter

To make a good approximation of the energy savings the $H_{tankhalf}$ will be used. For the flowrate the following percents per piers will be used:

- D-pier: 5,4 %
- E-pier: 17,4 %
- F-pier: 33,3 %
- G-pier: 26,2 %
- Centrum: 17,7 %

In appendix G2 to G9 the savings in kWh and guilders are calculated.

The peaksavings are divided into two parts:

1. $(P_{oudmotor} - P_{nieuwmotor}) \cdot$ price, see table 12.2;
2. contractual power savings.

The contract power savings are calculated as follows:

- Average increase a month: 89 kW (measured from energy bills 1996 and 1997);
- Power factor = 0,85;
- Costs extra kVA contractual power: *f* 50,-
- Contractual power savings:

$$((P_{new\ average\ power} - P_{new\ vfd}) \cdot 50,\text{-}) / (0,85) \qquad (12.5.2)$$

These contractual power savings are also given in table 12.2.

These two calculations are also performed for the next year. The result is given in table 13.3.

It is noticed that the savings of 4 VSP's to 7 VSP's are almost equal. The savings at the situation of 8 VSP's is due to the fact that the Q_{max} is taken equal to 2150 m^3/h. If the actual flowrate is bigger than this value the savings at 8 VSP's will be equal to the situation of 7 VSP's.

Table 12.2: Total peak savings for the different alternatives in the first year.

Situation	Peaksavings [*f* 1000,-]	Contractual power savings [*f* 1000,-]	Total Savings [*f* 1000,-]
1 VSP	3,2	1,3	4,5
2 VSP's	4,3	1,7	6
3 VSP's	4,8	1,9	6,7
4 VSP's to 7 VSP's	5,1	2	7,1
8 VSP's	15,6	6	21,6

Table 12.3: Total peak savings for the different alternatives in the second year.

Situation	Peaksavings [*f* 1000,-]	Contractual power savings [*f* 1000,-]	Total Savings [*f* 1000,-]
1 VSP	7,3	2,8	10,1
2 VSP's to 7 VSP's	7,4	2,9	10,3

A summary of all savings in the first year is given in table 12.4.

Table 12.4: Annual approximate savings (in 1000 guilders) for the different alternatives.

New Situation	Savings first year [*f* 1000,-]	Difference [*f* 1000,-]
1 VSP and m FSP's	20	
2 VSP's and m FSP's	35	15
3 VSP's and m FSP's	45	10
4 VSP's and m FSP's	53	8
5 VSP's and m FSP's	58	5
6 VSP's and m FSP's	60	2
7 VSP's and m FSP's	61	1
8 VSP's and **no** FSP's	75	14

It is noticed that:

- The savings for an extra VSP decreases rapidly;
- At the situation of 8 VSP's the flowrate peak should not exceed 8*288 m^3/h, because then the big difference caused by the power peak savings, will become less. When this happens the savings in relation to the savings of 7 VSP's will be almost zero.
- The Duty Cycle is calculated from the flow-time graph in February.
- The accuracy of these calculations is:
  - calculation of the pipeline/valves losses: 0,1%
  - efficiency: 0,1%
  - duty cycle: 1,0%

Now only the first four situations will be considered as an attractive project. From the financial analysis a decission can be made what the best solution will be (see chapter thirteen).

Chapter 13

Financial analysis

13.1 Introduction

VFD's can save a considerable amount of energy, but of course there is a significant expenditure of capital required to purchase and install the drive. Thus the calculations to arrive at the energy savings are only half the economic evaluation story. There must also be a means to take the guilder value of energy savings and compare it against the required initial investment in order to determine whether it is indeed an attractive project. There is a vast amount of literature available on various types of engineering economic analysis and virtually all companies will have their own accounting methods for doing this evaluation.

A simple payback calculation taking only power costs into account may be sufficient to determine project viability, based on some assumed figure for cents/kWh, as in equation 13.1.1:

$$simplepayback(yrs) = \frac{totalproject\cos ts(guilders)}{\frac{cents}{kWh} \cdot MWh \cdot 10} \qquad (13.1.1)$$

A more sophisticated financial analysis tool may be warranted. Now two widely accepted measurements of a project's economic viability will be discussed:

1. the simple time to payback (STP);
2. the discounted cash flow rate of return (DRR), which is also referred to as the return on investment (ROI).

Both calculations take into account the effect of tax rate, investment tax credit (if any), type and time period of depreciation, power cost, and any annual increase in the power cost. (These factors are what makes STP different from the simple payback). The big difference between the two calculations is that STP does not take into account the time value of money whereas the DRR does.

The STP is often used as a first screening in looking at project viability. This number is simply the time required to pay back from the energy savings a project's original cost.

The DRR is a more sophisticated measurement than the STP since it recognizes the time value of money. The relationship between investment, savings, and DRR is indicated by:

$$C = \sum_{n=1}^{n_{max}} \left[\frac{S}{(1+DRR)^n} \right] \tag{13.1.2}$$

where C = initial capital expenditure, in guilders
S = yearly savings, in guilders
n = year
n_{max} = length of evaluation period, in years
DRR = Discounted Rate of Return, in per unit

The DRR is simply the interest rate earned on an original investment over an assumed evaluation period of normally five to ten years. If the STP equals the evaluation period, the DRR is 0 percent, since only the project cost is recouped and no additional cash is generated to "pay interest".

The following method, given in table 13.1 will be used to calculate the payback period and the DRR:

Power cost =	**x** cents per kWh	
Anual power cost increase =	**y** %	
Depreciation method =	straight line	
Corporate tax rate =	**z** %	(normally 35%)
Investment tax rate =	**u** %	(normally 0%)
Depreciation period =	**v** year	

Table 13.1: Method to calculate the Cash flow analysis:

Year	Gross Savings (GS)	Tax Depreciation (TD)	Taxable Income (TI)	Income Taxes (IT)	Cash Flow (CF)	Cumultative Cash Flow (CCF)
0	- C	0	0	0	- C	- C
1	T_1	C / v	T_1 - C/v	(TI_1)*z/100	$GS_1 - IT_1$	CCF_0+CF_1
2	T_2	C / v	T_2 - C/v	(TI_2)*z/100	$GS_2 - IT_2$	CCF_1+CF_2
3	T_3	C / v	T_3 - C/v	(TI_3)*z/100	$GS_3 - IT_3$	CCF_2+CF_3

where $T_1 = S * x$
$T_2 = S * (x + x*y/100) = S * x_2$
$T_3 = S * (x_2 + x_2*y/100) = S * x_3$

STP = the year when the cummulative cash flow equals zero

The DRR over ten years (DRR_{10}) can be calculated as follows:

$$-C+\frac{(CCF_1)}{(1+DRR_{10})}+\frac{(CCF_2)}{(1+DRR_{10})^2}+...+\frac{(CCF_9)}{(1+DRR_{10})}+\frac{(CCF_{10})}{(1+DRR_{10})}=0 \qquad (13.1.3)$$

Normally a project which has a DRR of at least 12 % is considered to be valid.

13.2 Subsidies

At this moment it is very important, considering the environment, to produce less CO_2 emissions. This is the additional advantage when saving energy costs, because for every kWh of energy saved, 1 kg less of carbon dioxide is emitted to the atmosphere.

This reduction caused by the implementation of VFD's is considered as a project where subsidies are possible. In the Netherlands one can obtain subsidies at two different government companies. They are:
1. NOVEM (TIEB-regeling);
2. SENTER.
3. Investment tax credit at the 'belastingdienst'.

The first one can't be used, because they only give subsidies to projects which are completely new in the considered branch. Senter, however, still gives subsidies to projects such as the implementation of VFD's in connection with pumps (number 130602 in the Senter EIA brochure). This is called the 'Energie-Investerings-Aftrek' (EIA), which means that one can depreciate an extra amount of money. The relation between the investment costs and the subsidies are given in table 13.2.

Table 13.2: Relation investment costs and the different subsidies.

Investment costs [*f* 1000,-] >	Investment costs [*f* 1000,-] ≤	EIA [%]	Investment Tax credit [%]	Total [%]
3,6	61	52	25	76
61	120	50,5	22	72,5
120	179	49	19	68
179	238	47,5	16	63,5
238	297	46	13	59
297	356	44,5	11	55,5
356	416	43	8	51
416	475	41,5	5	46,5
475		40	0	41

As an extra subsidy one can use the SENTER 'VAMIL' project. This means that one can depreciate the amount of money when ever one wants. So one can equal the depreciation to the savings and one doesn't have to pay corporate tax rate in that year. The pump-VFD implementation is also a VAMIL project (number 6033 in the SENTER VAMIL brochure).

The last one is the investment tax credit. This is an extra subsidy, such as the EIA, and is also given in table 13.2.

13.3 Investment costs

After determining the expected savings, they can be used as the basis for a payback analysis. To do this, however, all required hardware and installation costs must be defined.

The following investment costs must be considered for the application at AFS:
- Replacement of the motor by an Ex(d) motor;
- Installation costs and price of the VFD;
- Cabels to connect the VFD with the motor;
- Installation of a airconditioning in the room where the VFD is situated;
- Pump control has to be renewed;
- Cooling requirements for the motor;
- Development of a software programme.

In Appendix H1 to H4 the investment costs for the four different situations are calculated. In the E&I room at Rijkl the maximum amount of VFD's that can be installed equals four. So this is also a reason that only these four situations are considered.

A summary of these investment costs are given in table 13.3:

Table 13.3: Investment costs for different situations.

Situation	Investment Costs [*f* 1000,-]
1 VSP	181
2 VSP's	294
3 VSP's	425
4 VSP's	536

Drawings of the new installations for these situations are given in Appendix I.

13.4 Calculation of the payback time and the DRR

The calculations of the cummulative cash flow for the four different situations are performed as explained in paragraph 13.1. The following conditions are fulfilled:

- The peak- and contract power savings from year three to year ten are an average of the first and second year.
- The annual power cost increase is: 3 %;
- An annual increase in kWh at Rijkl is: 6 % (obtained from the last five years);
- Depreciation method = straight line (for the no subsidies case);
- Depreciation period = 10 year;
- Corporate tax rate = 35 %;
- Subsidies of government (given in table 13.2).

The calculations are given in Appendix J1 to J8. The payback time and the DRR are given in table 13.4.

From table 13.4 it can be concluded, when a DRR of at least 12 % is considered, that the only two situations which can be implemented are:

- 1 VSP;
- 2 VSP's;

Table 13.4: STP and DRR

Situation	No Subsidies		Subsidies	
	STP [year]	DRR [%]	STP [year]	DRR [%]
1 VSP	7,2	8,9	6,4	**12,5**
2 VSP's	7,1	9,4	6,4	**12,9**
3 VSP's	7,7	7,1	7	9,8
4 VSP's	8	6	7,2	8,2

Chapter 14
Future Perspectives

14.1 Introduction

In this chapter some possible future changes and applications will be dealt with. These are the following:
- Flattening the fueling peaks;
- Direct pumping from RijkII into the 32 inch pipeline;
- Installation of a new pipeline from Amsterdam to Schiphol.

14.2 Flattening the fueling peaks

At the moment a lot of airplanes are tanked at the following periods of time:
1. 08.00 am till 10.00 am;
2. 12.00 am till 02.00 pm;

which cause the big peaks in power absorbed at RijkII (more flow required equals more pumping). In the future Schiphol plans to flatten these fueling peaks, so that an equal amount of flow is pumped over a bigger period of time.

The question that will rise is: will this affect a new implementation such as the VFD? Three different conditions will now be considered:
1. Flow rate is constant over 24 hours, which means that the Q is equal to 404 m^3/h;
2. Flow rate is constant over 16 hours, which means that the Q is equal to 606 m^3/h;
3. Flow rate is:
   - 0.00 am till 06.00 am: $Q = 0$ m^3/h
   - 06.00 am till 12.00 am: $Q = 600$ m^3/h;
   - 12.00 am till 03.00 pm: $Q = 888$ m^3/h;
   - 03.00 pm till 09.30 pm: $Q = 450$ m^3/h;
   - 09.30 pm till 0.00 am: $Q = 200$ m^3/h.

The thirth one is the most realistic one.

The same calculations as in chapter twelve are performed:
- A FSP pumps 288 m^3/h, which equals a pump head of 122,1 meter;
- $P_{VSP}(288) = 106{,}2$ kW;
- $n \cdot Q_{VSP} = Q_{tot} - m \cdot Q_{FSP}$;
- $\eta_{mo} \cdot \eta_{co} = 0{,}95$.

In Appendix K the calculations are performed. A brief summary is given in table 14.1.

Table 14.1: Savings when Schiphol flattens the peaks.

Situation	Annual VFD Savings [*f* 1000,-]	Annual Peaksavings with/without VFD's [*f* 1000,-]
1. 24 hours constant flow		100
1 VSP	21	
2 VSP's	74	
2. 16 hours constant flow		89
1 VSP	18	
2 VSP's	26	
3 VSP's	77	
3. Parts constant		72
1 VSP	20	
2 VSP's	40	
3 VSP's	61	
4 VSP's	83	

In table 14.1 also the annual peaksavings, that will happen, when Schiphol flattens the flow, are given. These are calculated for the actual throughput at Schiphol. Of course for this savings it doesn't matter if VFD's are applied.

From table 14.1 it can be concluded that the annual savings are the same or in some cases even bigger than the savings obtained in chapter twelve. So when it is decided to install VFD's one doesn't have to worry that the savings will be less when Schiphol flattens their peaks.

14.3 Direct pumping from RijkII into the 32 inch pipeline

AFS is planning to connect the tanks at RijkIIA directly to the 32 inch pipeline. The pumps at RijkII which at this moment are used as booster pumps to pump from RijkIIA to RijkIIB, will be used as the new hydrant pumps.

When the pumps at RijkII are no longer used as booster pumps, this pumping will be saved. This results in:

0,9 (percent pumping systems at RijkII) $\cdot f$ 163.000,- (energy bill 1996) =
***f* 147.000,- saved.**

14.4 Amsterdam Schiphol Pipeline

AFS wants to install a new pipeline from Amsterdam to Schiphol, the Amsterdam-Schiphol Pipeline (ASP). When this pipe is ready no longer jetties need to deliver the kerosine at Oost.

When the pump head at Amsterdam is that big, that it will be able to pump the kerosene from Amsterdam to Schiphol in one time, the pumps at Oost no longer have to be used. This will result in:

0,75 (percent pumping at Oost) $\cdot f$ 146.000,- (energy bill 1996) =
***f* 110.000,- saved.**

It can be concluded that the future perspectives mentioned in paragraph 14.3 and 14.4 are good investments.

Chapter 15

Conclusions and recommendations

There is probably not a pump station that could not realize some energy cost savings. So can AFS. Before investing in some sophisticated optimization software, one should first conduct an overall evaluation of the energy costs associated with pumping to determine whether there is much room for improvement in pump operation. For most utilities, the savings realized from such an evaluation can quickly pay for the evaluation. This also was the case at AFS:

By reviewing the energy bills, it can be concluded that:

- The pumping systems were the main energy users: about 85%;
- AFS will have to raise its contractual power limit at RijkI;
- AFS paid too much at RijkII in 1996: about ***f* 20.000,-**
- AFS paid too much at Oost due to the fact that the costs lag behind the actual throughput: about ***f* 17.000,-**

When one wants to optimalize a pump system, one should always keep in mind the policy of the company. At AFS the main objective is the demand at Schiphol must be satisfied. So the flow rate can't be changed. An other solution is decreasing the head.

By checking the head produced by the pumps and the head required at Schiphol, it turned out, that especially at the low flow rates, the head produced was much bigger than the head delivered. So energy savings are possible if the head produced is decreased.

By comparing some different Variable Speed Drives, the *best* solution with respect to rangeability, speed of response, control accuracy, and maintenance is the Variable Frequency Drive (VFD).

The evaluation of the savings possible by the use of VFD's is not necessarily a straightforward simple procedure. Considerable process and drive system data must be obtained to ensure a correct evaluation. The method that best insures that all required factors will be considered is to calculate seperately the energy usage by the VFD system and the base comparison system, and then evaluate the energy savings between the two figures.

After producing such a realistic energy evaluation, it turned out that by comparing different situations, the following ones were attractive:

- 1 Variable Speed Pump (VSP);
- 2 VSP's
- 3 VSP's
- 4 VSP's

Also if one considers the placement of the VFD's, a maximum amount of 4 VFD's could be installed at this moment, without having to build a new building.

After determining the savings, they can be used as the basis for a payback analysis. To do this, however, the investment costs should be defined. By taking into account

- these investment costs;
- the annual increase in power cost
- the different taxes;
- the depreciation method;
- government subsidies (SENTER);

the payback period and the Discounted cash flow Rate of Return (DRR) can be calculated for the four situations mentioned above. It turned out, by considering a DRR $\geq$ 12 %, only the following situations were attractive:

- 1 VSP with subsidies
  - Payback period: 6,4 year
  - DRR: 12,5 %
- 2 VSP's with subsidies:
  - Payback period: 6,4 year
  - DRR: 12,9 %

By keeping in mind that a spare system is important, the **best** solution is the situation with 2 VSP's with government subsidies.

Of course for AFS it is very important to keep in mind the future perspectives. The following can be concluded:

1. If Schiphol flattens the fueling peaks and VSP's are installed, this will not result in a decrease in savings, but in an **increase**;
2. If AFS pumps directly from RijkII into the 32 inch pipeline by using the pumps at RijkII, this will result in an annual saving of ***f* 147.000,-**
3. If AFS finishes the Amsterdam-Schiphol Pipeline and the head produced at Amsterdam will be that big that it can pump the kerosene in one time, the savings are at Oost: ***f* 110.000,-**

By taking into account all the information given above and some additional advantages, such as:

- Soft start;
- Reduction of CO_2 emissions;
- Reduction of inrush current,

it is recommended to invest in two VSP's.

If AFS decides to install two VSP's, the following approach could serve as a reasonable guideline toward assuring trouble free reliability:

1. Select the VFD primarily based on application. A common error is to buy strictly on price - without proper application considerations.
2. Address the issue of applying the correct commutating reactance.
3. Review/evaluate vendors' drives
4. Address electrical noise - especially for numerous drives on a common bus.
5. Where a significant portion of the load is comprised of VFD's, perform a power factor and harmonic study.
6. Address grounding issues.
7. If AFS has a history of unexplained nuisance tripping or drive failures, an in-depth power quality investigation should be performed.

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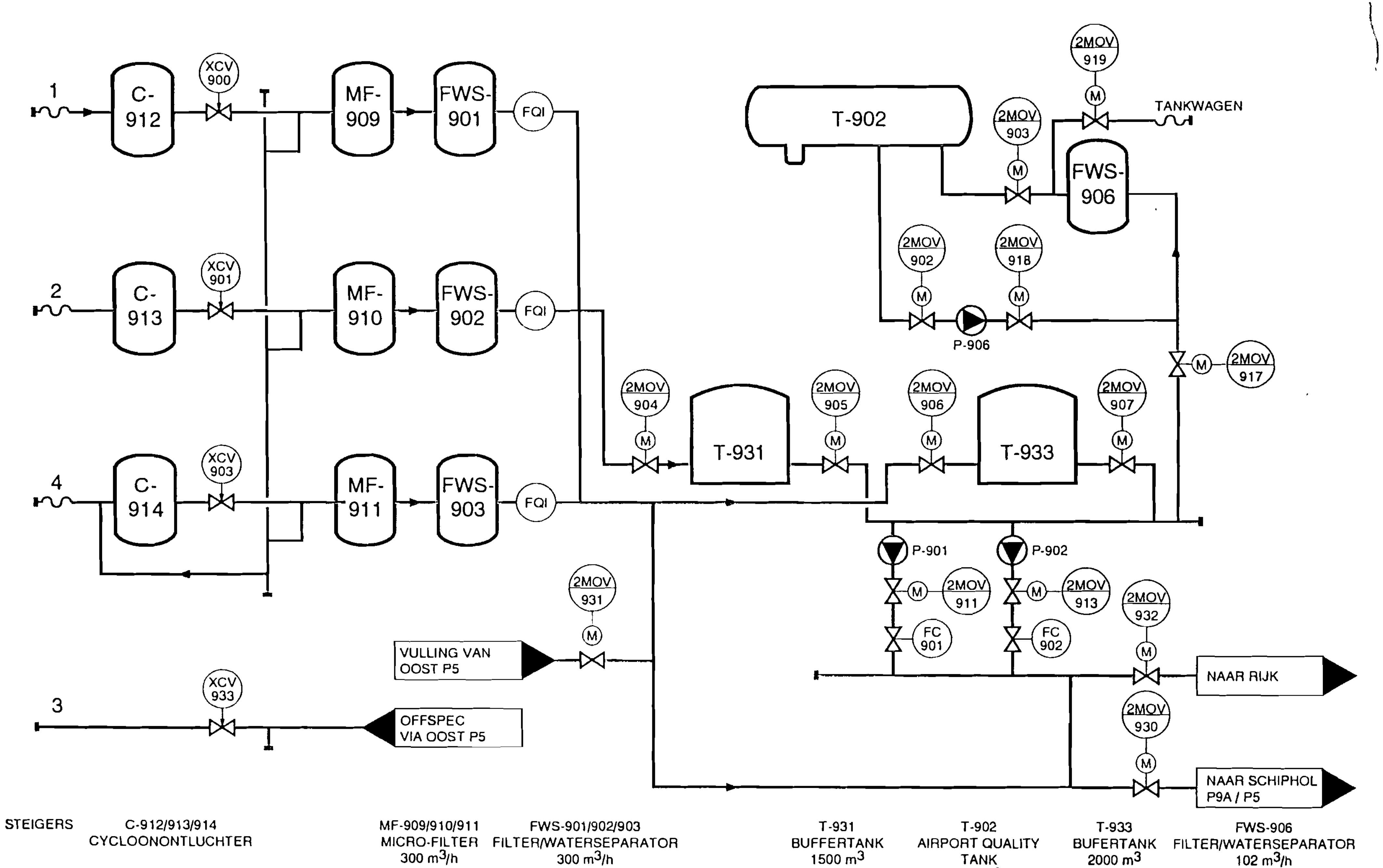
Appendix A1

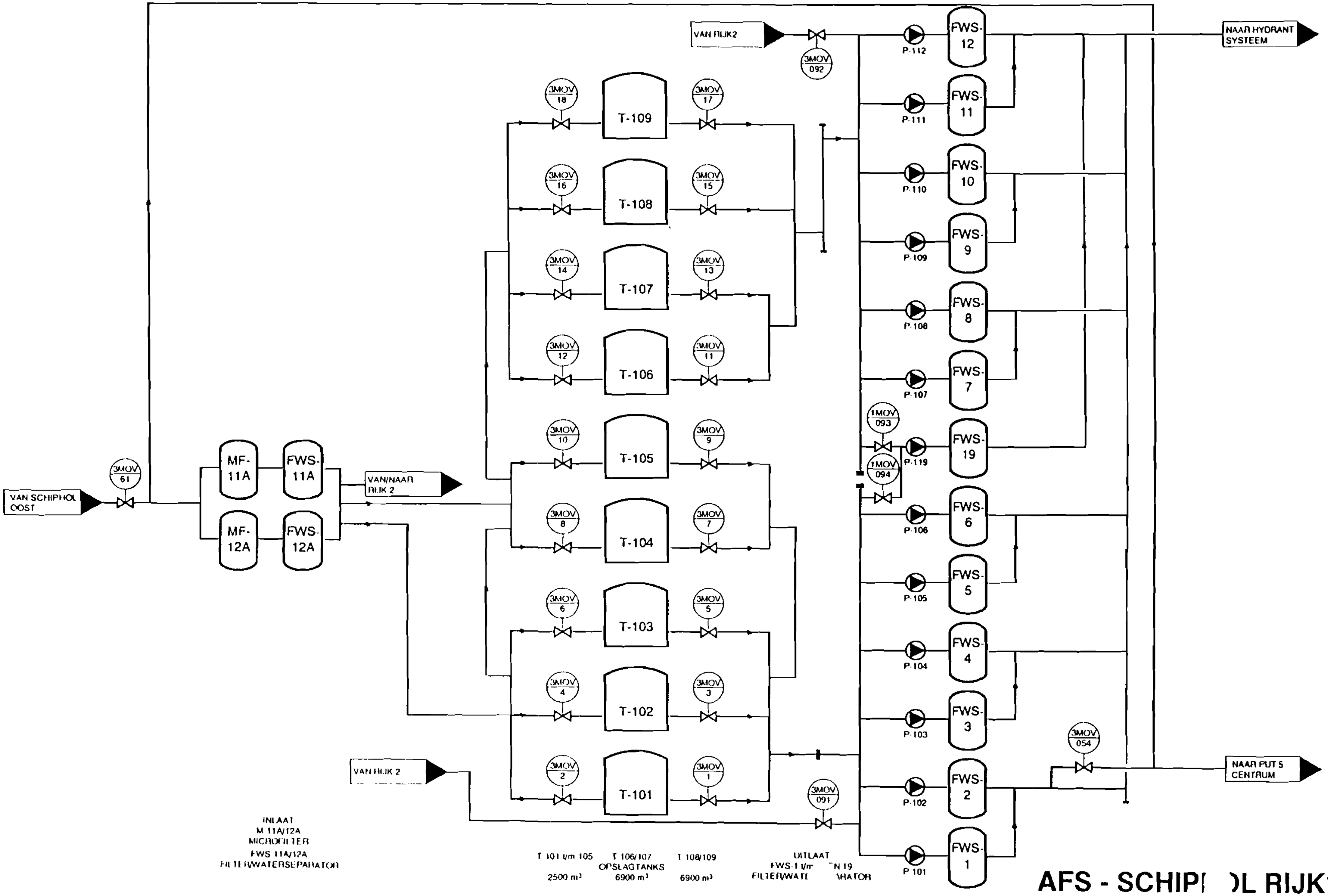
AFS Schiphol Oost.



AFS - SCHIPHOL OOST

Appendix A2

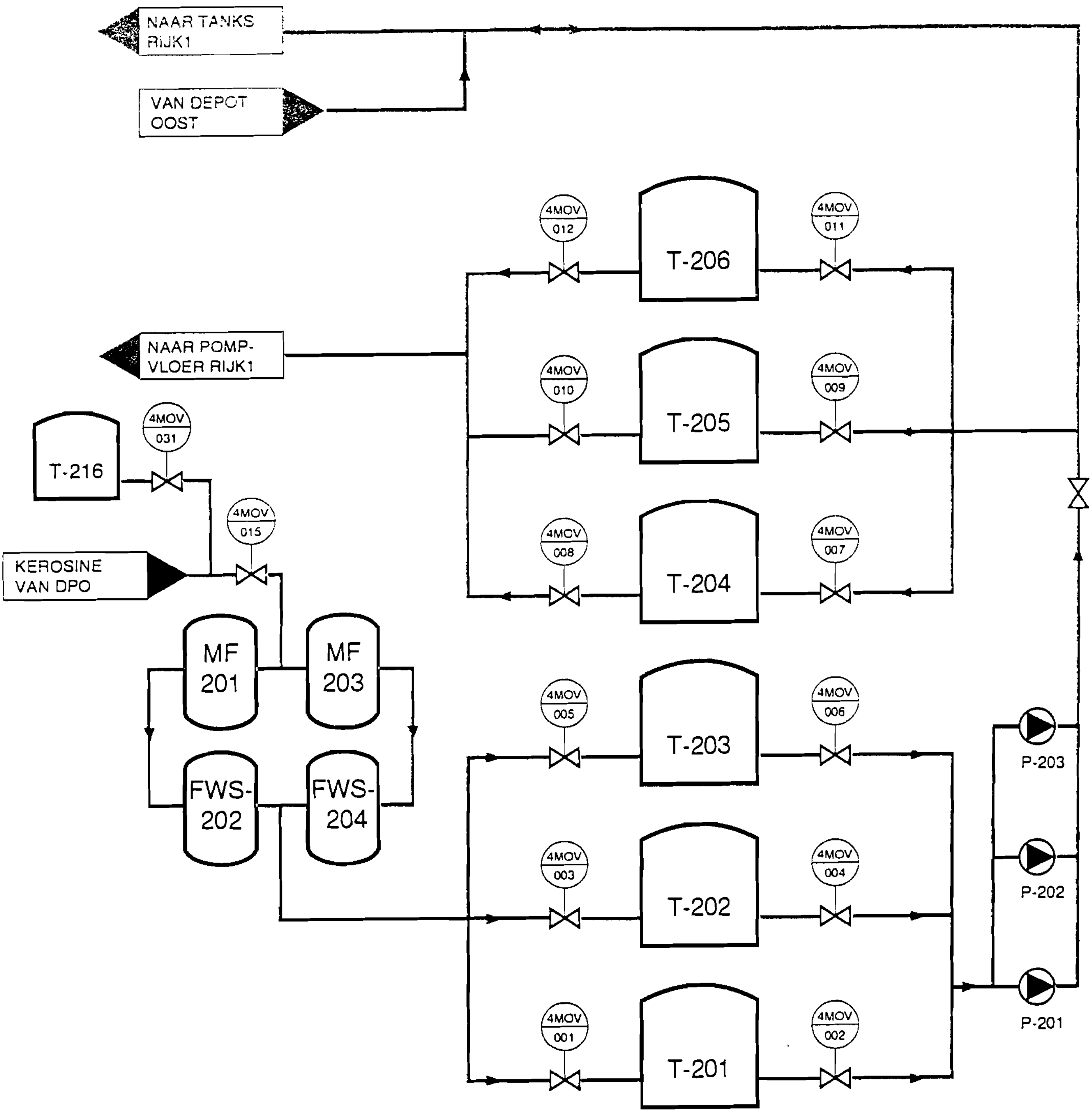
AFS Schiphol RijkI.



AFS - SCHIP[ )L RIJK1

Appendix A3

AFS Schiphol RijkII.



MF-201/203
MICROFILTER

FWS-202/204
FILTER/WATERSEPARATOR

T-201 t/m 206
OPSLAGTANKS
12434 m^3

AFS - SCHIPHOL RIJK2

Appendix B

AFS Energy bills 1996.

Aircraft Fue. ..ply B.V.

KOSTEN VAN ELECTRICITEIT

Contractnummer	periode	Verbruik (KWh)	Vermogen (KW)	Energiekosten (Hfl)	(Hfl/KWh)	Brandstofkosten (Hfl)	(Hfl/KWh)	Kosten vermogen (Hfl)	(Hfl/KW)	Kn. electrische werken (Hfl)	Vaste kosten (Hfl)	Totaal 1996	Totaal 1995
Electriciteit depot Oost													
NGV 90 AFS 01	januari	75.000	270	1.875,00	0,0250	4.207,50	0,0561	9.234,00	34,2000	1.606,00	90,00	17.012,50	19.840,00
(Grootverbruik)	februari	54.000	282	1.350,00	0,0250	3.029,40	0,0561	9.644,40	34,2000	1.606,00	90,00	15.719,80	18.571,60
(depot Oost)	maart	72.000	264	1.800,00	0,0250	4.039,20	0,0561	4.514,40	17,1000	1.606,00	90,00	12.049,60	14.325,40
	april	51.000	252	1.275,00	0,0250	2.728,50	0,0535	4.309,20	17,1000	1.606,00	90,00	10.008,70	16.039,00
	mei	57.000	258	1.425,00	0,0250	3.049,50	0,0535	4.411,80	17,1000	1.606,00	90,00	10.582,30	15.539,20
	juni	45.990	255	1.149,75	0,0250	2.460,47	0,0535	4.360,50	17,1000	1.606,00	90,00	9.666,72	16.692,10
	juli	49.280	252	1.232,00	0,0250	2.636,48	0,0535	4.309,20	17,1000	1.606,00	90,00	9.873,68	14.132,20
	augustus	49.290	258	1.232,25	0,0250	2.637,02	0,0535	4.411,80	17,1000	1.606,00	90,00	9.977,07	13.157,80
	september	45.380	262	1.134,50	0,0250	2.427,83	0,0535	4.480,20	17,1000	1.606,00	90,00	9.738,53	13.304,80
	oktober	47.630	262	1.190,75	0,0250	2.600,60	0,0546	4.480,20	17,1000	1.606,00	90,00	9.967,55	12.036,40
	november	58.670	276	1.466,75	0,0250	3.203,38	0,0546	9.439,20	34,2000	1.606,00	90,00	15.805,33	16.043,20
	december	57.420	280	1.435,50	0,0250	3.135,13	0,0546	9.576,00	34,2000	1.606,00	90,00	15.842,63	18.084,40
	CBS	**662.660**	3.171	16.566,50	0,0250	36.155,01	0,0546	73.170,90	23,0750	19.272,00	1.080,00	146.244,41	187.766,10
												0,00	0,00
											Totaal depot Oost	**146.244,41**	**187.766,10**
Electriciteit depot Rijk 1													
NGV 87P-R01 A	januari	111.357	744	7.572	0,0680	6.247	0,0561	14.285	19,2000	4.115	90	32.309,21	30.777,04
(Grootverbruik)	februari	113.183	802	7.696	0,0680	6.350	0,0561	15.398	19,2000	4.115	90	33.649,41	37.366,22
(depot Rijk 1)	maart	115.532	825	7.856	0,0680	6.481	0,0561	7.920	9,6000	4.115	90	26.462,53	25.179,54
	april	130.244	888	8.857	0,0680	6.968	0,0535	8.525	9,6000	4.115	90	28.554,44	27.511,44
	mei	141.962	899	9.653	0,0680	7.595	0,0535	8.630	9,6000	4.115	90	30.083,79	21.288,27
	juni	128.683	942	8.750	0,0680	6.885	0,0535	9.043	9,6000	4.115	90	28.883,18	30.499,69
	juli	151.692	943	10.315	0,0680	8.116	0,0535	9.053	9,6000	4.115	90	31.688,38	24.649,91
	augustus	146.124	1.042	9.936	0,0680	7.818	0,0535	10.003	9,6000	4.115	90	31.962,26	29.663,70
	september	115.659	902	7.865	0,0680	6.188	0,0535	8.659	9,6000	4.115	90	26.916,77	29.237,90
	oktober	150.350	825	10.224	0,0680	8.209	0,0546	7.920	9,6000	4.115	90	30.557,91	29.799,48
	november	139.027	863	9.454	0,0680	7.591	0,0546	16.570	19,2000	4.115	90	37.819,31	33.469,35
	december	147.224	789	10.011	0,0680	8.038	0,0546	15.149	19,2000	4.115	90	37.403,46	37.543,40
	CBS	**1.591.037**	10.464	108.190,52	0,0680	86.484,93	0,0544	131.155,20	12,5339	49.380,00	1.080,00	376.290,65	356.985,94
											Ophoging contract i.v.m. verbruik 1995	0,00	7.700,00
											Totaal depot Rijk	**376.290,65**	**364.685,94**
Electriciteit depot Centrum													
NGV 87 TDT 01	januari	9.774	60	469	0,0480	564	0,0577	1.812	30,2000	56	90	2.991,11	4.311,73
(Grootverbruik)	februari	12.154	60	583	0,0480	701	0,0577	1.812	30,2000	56	90	3.242,68	4.027,59
(depot Centrum)	maart	11.474	60	551	0,0480	662	0,0577	906	15,1000	56	90	2.264,80	3.134,11
	april	11.274	60	541	0,0480	620	0,0550	906	15,1000	56	90	2.213,22	3.666,57
	mei	11.994	60	576	0,0480	660	0,0550	906	15,1000	56	90	2.287,28	3.459,01
	juni	18.514	60	889	0,0480	1.018	0,0550	906	15,1000	56	90	2.958,94	3.401,82
	juli	-166	60	-8	0,0480	-9	0,0550	906	15,1000	56	90	1.034,90	2.404,06
	augustus	11.754	60	564	0,0480	646	0,0550	906	15,1000	56	90	2.262,66	2.559,61
	september	11.874	60	570	0,0480	653	0,0550	906	15,1000	56	90	2.275,02	2.267,18
	oktober	5.274	60	253	0,0480	296	0,0561	906	15,1000	56	90	1.601,02	2.190,44
	november	10.400	60	499	0,0480	583	0,0561	906	15,1000	56	90	2.134,64	3.355,62
	december	36.668	71	1.760	0,0480	2.057	0,0561	2.144	30,2000	797	90	6.848,33	2.850,91
	CBS	**150.988**	731	7.247,40	0,0480	8.452,00	0,0560	13.922,20	19,0454	1.413,00	1.080,00	32.114,60	37.628,65
											Kosten water	4.121,02	7.808,93
											Kosten water dec 1995	2.960,12	0,00
											Kosten gas/gasolie	10.113,39	0,00
											Totaal depot Centrum	**49.309,13**	**45.437,58**

J.J. Heeringa

-0248

KOSTEN VAN ELECTRICITEIT

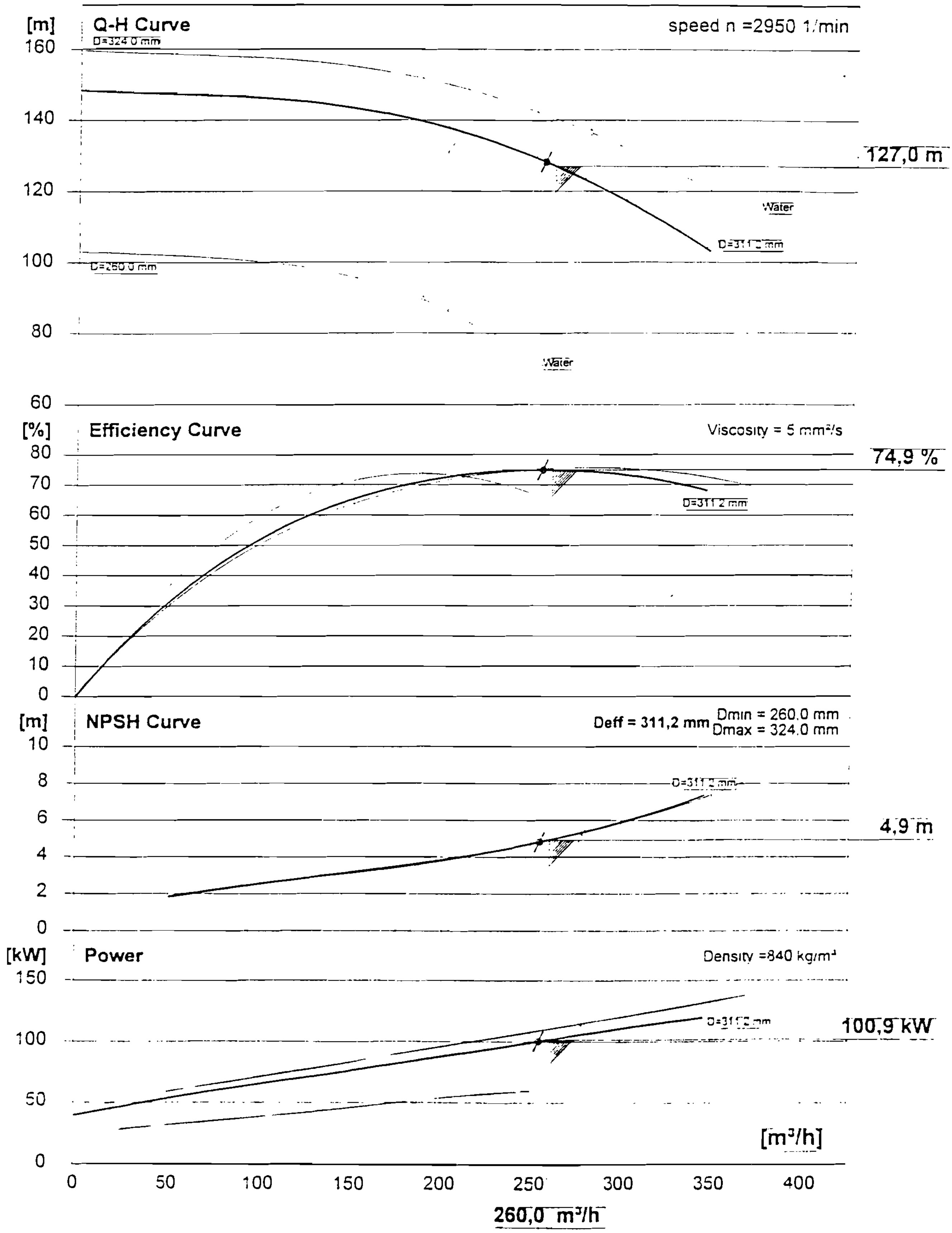
Contractnummer	periode	Verbruik (KWh)	Vermogen (KW)	Energiekosten (Hfl)	(Hfl/KWh)	Brandstofkosten (Hfl)	(Hfl/KWh)	Kosten vermogen (Hfl)	(Hfl/KW)	Kn. electrische werken (Hfl)	Vaste kosten (Hfl)	Totaal 1996	Totaal 1995
Electriciteit hydranten en pijpleidingen													
TKV 87F78900	januari					0		261,62		0	0	261,62	911,80
(kleinverbruik)	februari					0		1.210,40		0	0	1.210,40	817,91
(Hydrant)	maart					0		349,51		0	0	349,51	826,45
	april					0		390,49		0	0	390,49	747,79
	mei					0		260,74		0	0	260,74	721,85
	juni					0		260,74		0	0	260,74	921,63
	juli					0		260,74		0	0	260,74	597,47
	augustus					0		260,74		0	0	260,74	699,89
	september					0		260,74		0	0	260,74	708,43
	oktober					0		261,10		0	0	261,10	716,96
	november					0		261,10		0	0	261,10	665,75
	december					0		4.704,89		0	0	4.704,89	1.041,29
				0,00		0,00		8.742,81		0,00	0,00	8.742,81	9.377,22
												0,00	0,00
											Totaal hydranten en pijpleidingen	**8.742,81**	**9.377,22**
Electriciteit depot Rijk 2(rek.nr.)													
NGV RIJK 2 01 A	januari	55.500	294	2.220	0,0400	3.114	0,0561	6.642	22,5918	2.142	90	14.207,55	0,00
(Grootverbruik)	februari	73.750	294	2.950	0,0400	4.137	0,0561	6.791	23,0969	2.142	90	16.109,88	0,00
(depot Rijk)	maart	67.580	294	2.703	0,0400	3.791	0,0561	3.969	13,5000	2.142	90	12.695,44	0,00
	april	66.780	294	2.671	0,0400	3.573	0,0535	3.969	13,5000	2.142	90	12.444,93	0,00
	mei	70.270	294	2.811	0,0400	3.759	0,0535	3.969	13,5000	2.142	90	12.771,25	0,00
	juni	58.970	294	2.359	0,0400	3.155	0,0535	3.969	13,5000	2.142	90	11.714,70	0,00
	juli	70.190	294	2.808	0,0400	3.755	0,0535	3.969	13,5000	2.142	90	12.763,77	7.412,79
	augustus	70.090	294	2.804	0,0400	3.750	0,0535	3.969	13,5000	2.142	90	12.754,42	8.420,71
	september	55.440	294	2.218	0,0400	2.966	0,0535	3.969	13,5000	2.142	90	11.384,64	8.703,09
	oktober	76.220	294	3.049	0,0400	4.162	0,0546	3.969	13,5000	2.142	90	13.411,41	9.125,24
	november	74.600	294	2.984	0,0400	4.073	0,0546	6.764	23,0051	2.142	90	16.052,66	12.847,96
	december	87.560	294	3.502	0,0400	4.781	0,0546	6.656	22,6378	2.142	90	17.170,68	14.969,04
	CBS	**826.950**	3.528	33.078,00	0,0400	45.015,83	0,0544	58.603,50	16,6110	25.704,00	1.080,00	163.481,33	61.478,83
												0,00	0,00
											Totaal depot Rijk	**163.481,33**	**61.478,83**
												744.068,33	**668.745,67**
												=========	=========

Bron: facturenmappen.

Appendix C

Sulzer CZ 100-315 performance charts.

CZ 100-315

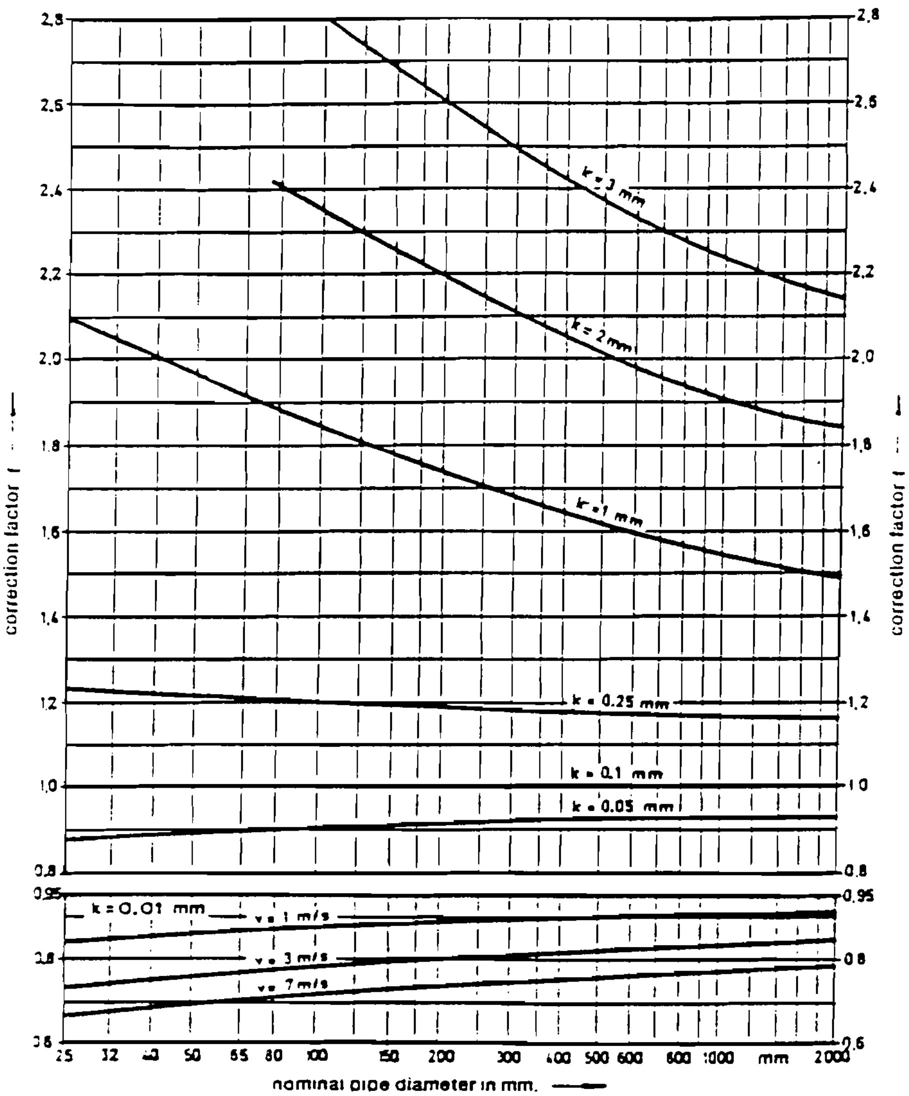

[m]
Q-H Curve
speed n =2950 1/min
160
140
120
100
80
60
127,0 m
Water
[%]
Efficiency Curve
Viscosity = 5 mm²/s
80
70
60
50
40
30
20
10
0
74,9 %
[m]
NPSH Curve
Deff = 311,2 mm
Dmin = 260,0 mm
Dmax = 324,0 mm
10
8
6
4
2
0
4,9 m
[kW]
Power
Density =840 kg/m³
150
100
50
0
100,9 kW
[m³/h]
0
50
100
150
200
250
300
350
400
260,0 m³/h

Appendix D

Tables piping systems

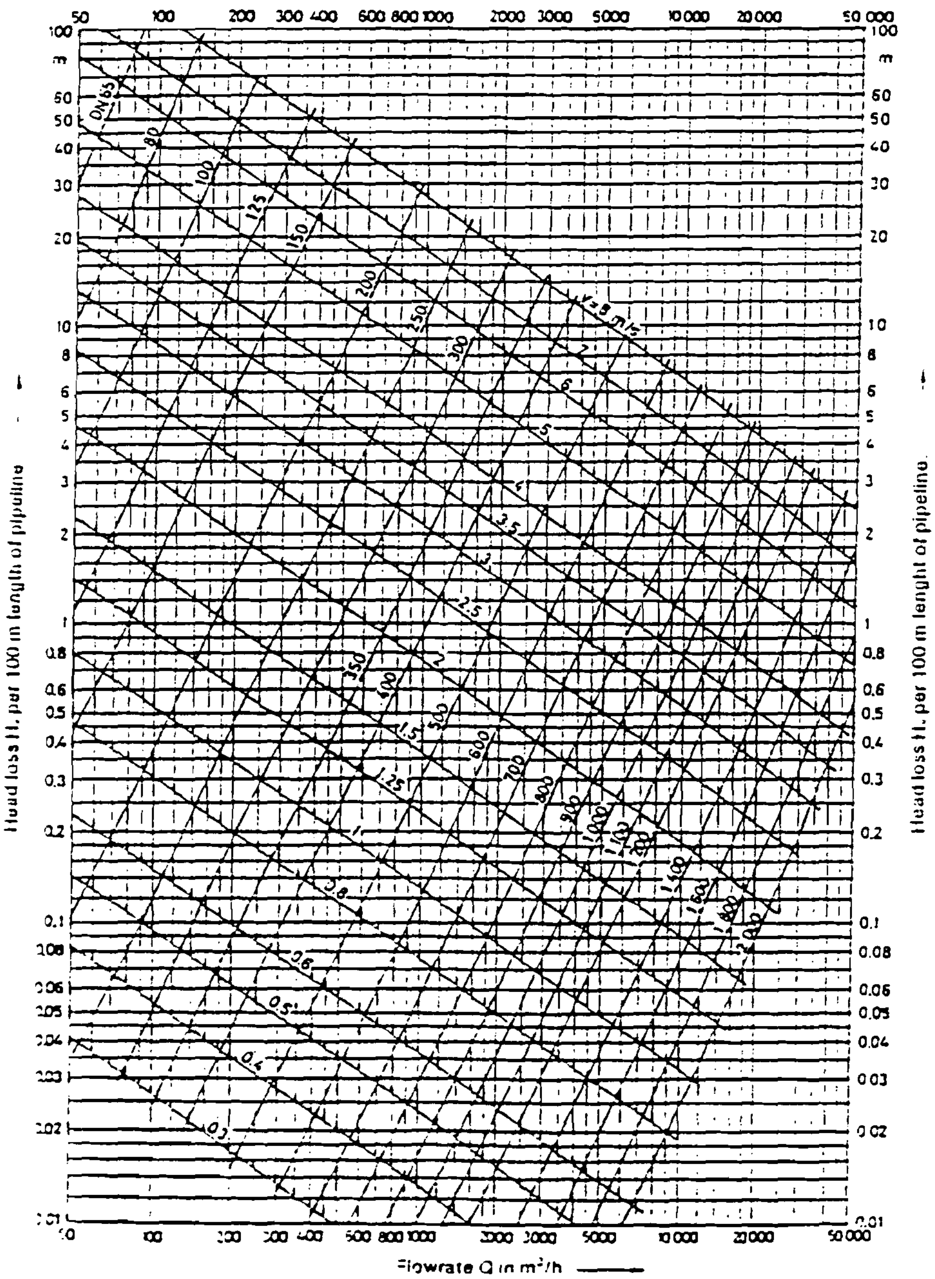
Appendix D1

Correction factors for roughness k ≠ 0,1 mm.



Appendix D2

Head loss H_v per 100m length of pipeline using Prandtl-Colebrook formula with k = 0,1 mm.



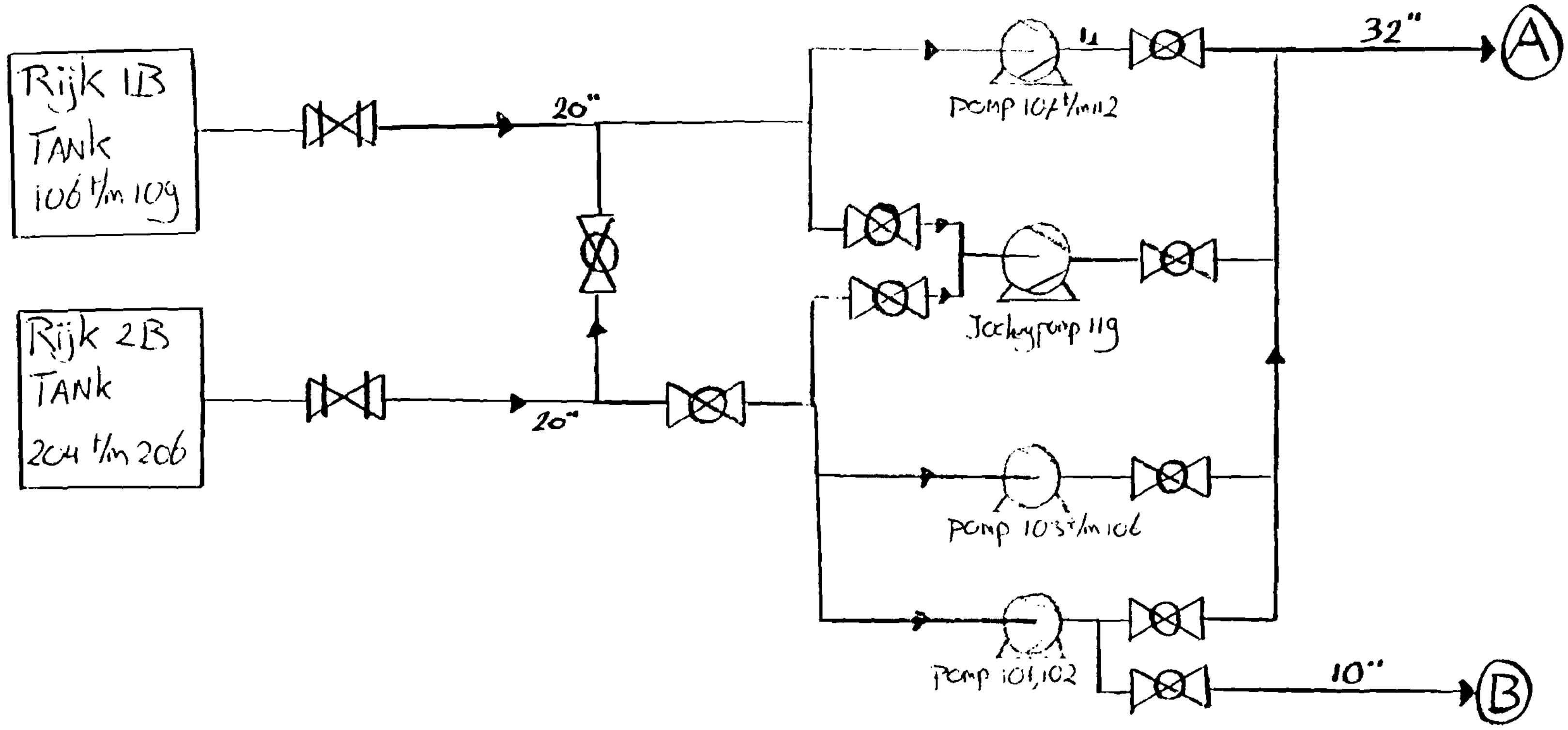
Appendix D3

Unit head loss in meters per one linear meter of pipe as function of diameter and the mean velocity; (New pipes:k=0,1mm en Existing pipes k=2mm).

Mean velocity	Pipe diameter 0.800 m — Pipe section 0.50265 m^2 — Head loss m/1 m of pipe length: New pipes	Existing pipes	Flow dm^3/s	Pipe diameter 0.900 m — Pipe section 0.63617 m^2 — Head loss m/1 m of pipe length: New pipes	Existing pipes	Flow dm^3/s
0.01			5.0205			6.3617
0.05	0.000004	0.000005	25.1328	0.000004	0.000005	31.8087
0.10	0.000014	0.000018	50.2656	0.000012	0.000015	63.6174
0.15	0.000029	0.000039	75.3984	0.000025	0.000034	95.4261
0.20	0.000049	0.000067	100.5312	0.000043	0.000058	127.2348
0.25	0.000074	0.000103	125.6640	0.000064	0.000087	159.0435
0.30	0.000103	0.000147	150.7968	0.000089	0.000124	190.8522
0.35	0.000137	0.000198	175.9296	0.000167	0.000167	222.6609
0.40	0.000174	0.000258	201.0624	0.000150	0.000218	254.4696
0.45	0.000216	0.000324	226.1952	0.000186	0.000274	286.2783
0.50	0.000262	0.000398	251.3280	0.000225	0.000336	318.0870
0.55	0.000312	0.000481	276.4608	0.000268	0.000406	349.8957
0.60	0.000367	0.000572	301.5936	0.000316	0.000483	381.7044
0.65	0.000425	0.000670	326.7264	0.000367	0.000565	413.5131
0.70	0.000489	0.000776	351.8592	0.000421	0.000654	445.3218
0.75	0.000557	0.000890	376.9920	0.000479	0.000749	477.1305
0.80	0.000628	0.001012	402.1248	0.000540	0.000852	508.9392
0.85	0.000703	0.001142	427.2576	0.000605	0.000961	540.7479
0.90	0.000781	0.001279	452.3904	0.000671	0.001077	572.5566
0.95	0.000864	0.001425	477.5232	0.000743	0.001199	604.3653
1.00	0.000952	0.001579	502.6560	0.000817	0.001327	636.1740
1.05	0.001044	0.001741	527.7888	0.000896	0.001461	667.9827
1.10	0.001139	0.001910	552.9216	0.000980	0.001606	699.7914
1.15	0.001239	0.002088	578.0544	0.001065	0.001752	731.6001
1.20	0.001341	0.002274	603.1872	0.001144	0.001910	763.4088
1.25	0.001448	0.002467	628.3200	0.001237	0.002073	795.2175
1.30	0.001559	0.002668	653.4528	0.001332	0.002241	827.0262
1.35	0.001673	0.002877	678.5856	0.001428	0.002420	858.8349
1.40	0.001791	0.003095	703.7184	0.001529	0.002604	890.6436
1.45	0.001914	0.003319	728.8512	0.001632	0.002737	922.4523
1.50	0.002041	0.003552	753.9840	0.001741	0.002983	954.2610
1.55	0.002174	0.003793	779.1168	0.001857	0.003186	986.0697
1.60	0.002309	0.004042	804.2496	0.001968	0.003398	1 017.8784
1.65	0.002449	0.004298	829.3824	0.002086	0.003610	1 049.6871
1.70	0.002593	0.004563	854.5152	0.002208	0.003837	1 081.4958
1.75	0.002740	0.004825	879.6480	0.002337	0.004061	1 113.3045
1.80	0.002890	0.005115	904.7808	0.002461	0.004299	1 145.1132
1.85	0.003044	0.005403	926.9136	0.002594	0.004538	1 176.9219
1.90	0.003202	0.005699	955.0464	0.002726	0.004792	1 208.7306
1.95	0.003363	0.006003	980.1792	0.002862	0.005044	1 240.5393
2.00	0.003530	0.006315	1 005.3120	0.003001	0.005307	1 272.3480
2.05	0.003700	0.006635	1 030.4448	0.003144	0.005578	1 304.1567
2.10	0.003875	0.006963	1 055.5776	0.003296	0.005850	1 335.9654
2.15	0.004052	0.007298	1 080.7104	0.003445	0.006136	1 367.7741
2.20	0.004234	0.007641	1 105.8432	0.003598	0.006424	1 399.5828
2.25	0.004419	0.007993	1 130.9760	0.003757	0.006712	1 431.3915
2.30	0.004611	0.008352	1 156.1088	0.003915	0.007025	1 463.2002
2.35	0.004806	0.008719	1 181.2416	0.004074	0.007319	1 495.0089
2.40	0.005006	0.009094	1 206.3744	0.004240	0.007641	1 526.8176
2.45	0.005209	0.009477	1 231.5072	0.004419	0.007960	1 558.6263
2.50	0.005416	0.009867	1 256.6400	0.004590	0.008288	1 590.4425
3.00	0.007695	0.014209	1 507.968	0.006518	0.011922	1 908.5220
3.50	0.010357	0.019340	1 759.296	0.008782	0.016246	2 226.6090
4.00	0.013405	0.025261	2 010.624	0.011367	0.021204	2 544.6960

Appendix E1

AFS piping system RijkI to hydrant system.



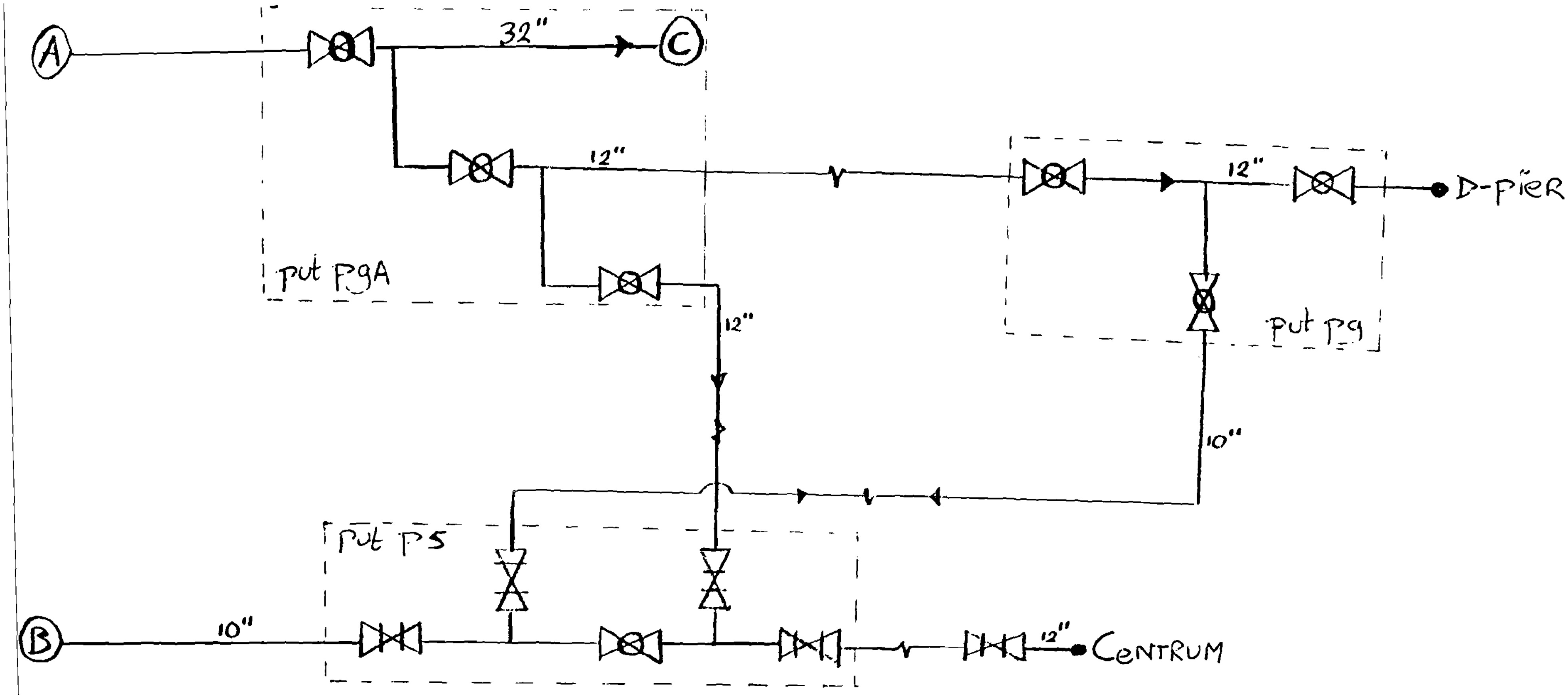
pompkamer Rijk

(A) gaat naar put P9A

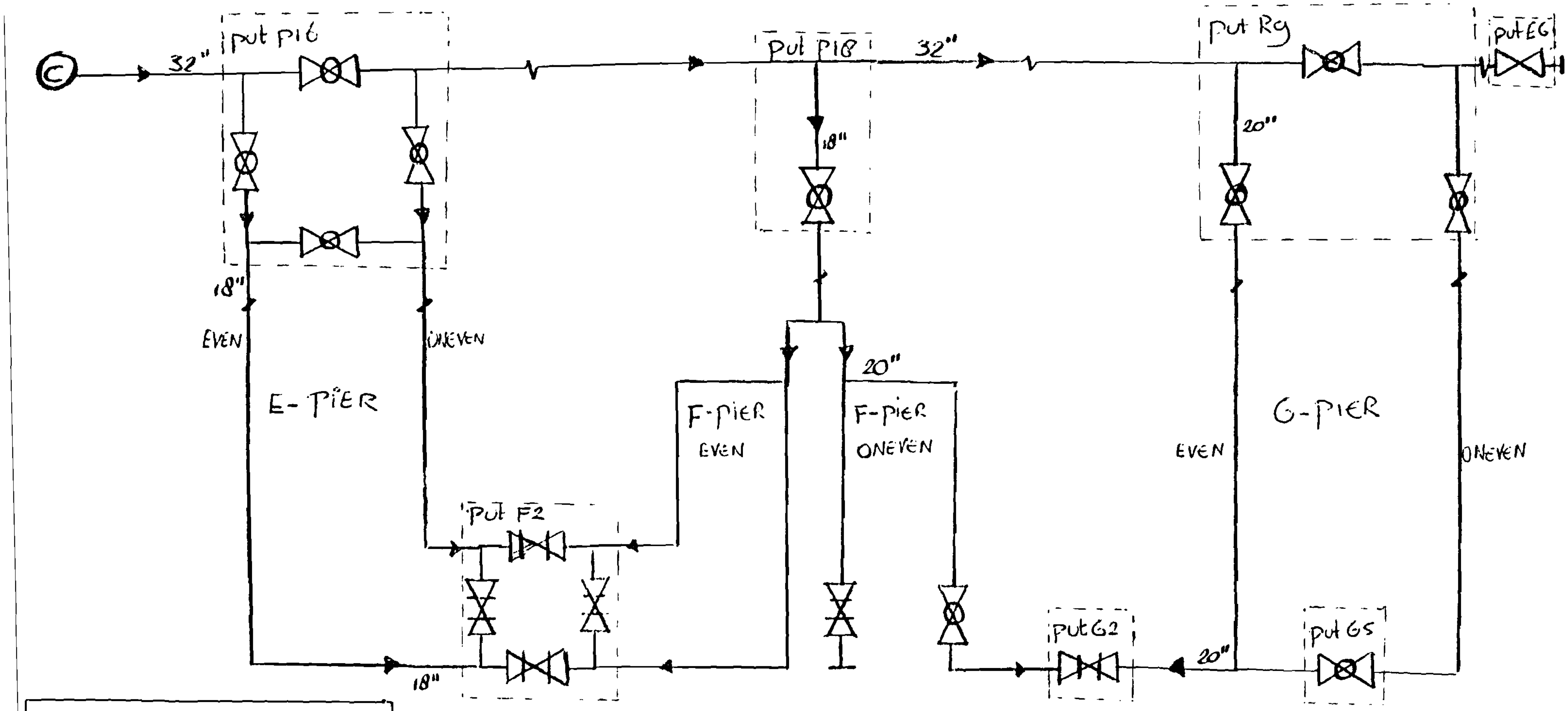
(B) gaat naar put P5

TWIN seal 'Double Block & Bleed' valve

Tk BALL valve



put P9A
put P9
put P5
D-pier
centrum (octaanplein)

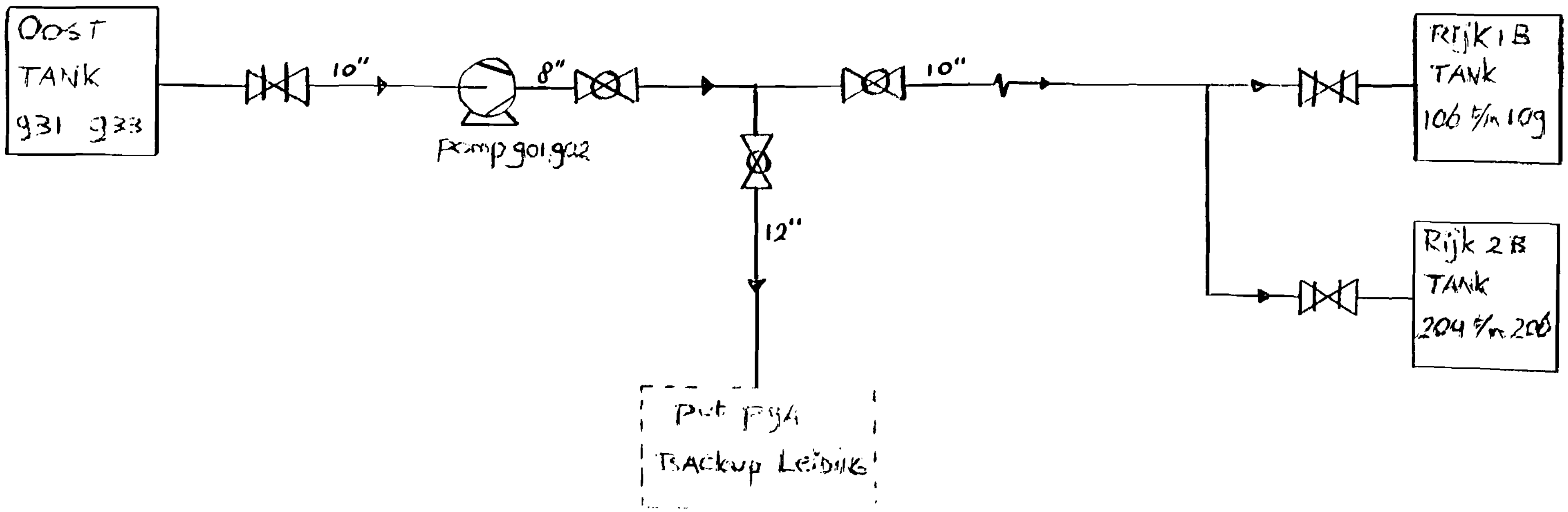


put p16
put p18
put R9
put EG
put F2
put G2
put G5

E-pier
F-pier
G-pier

Appendix E2

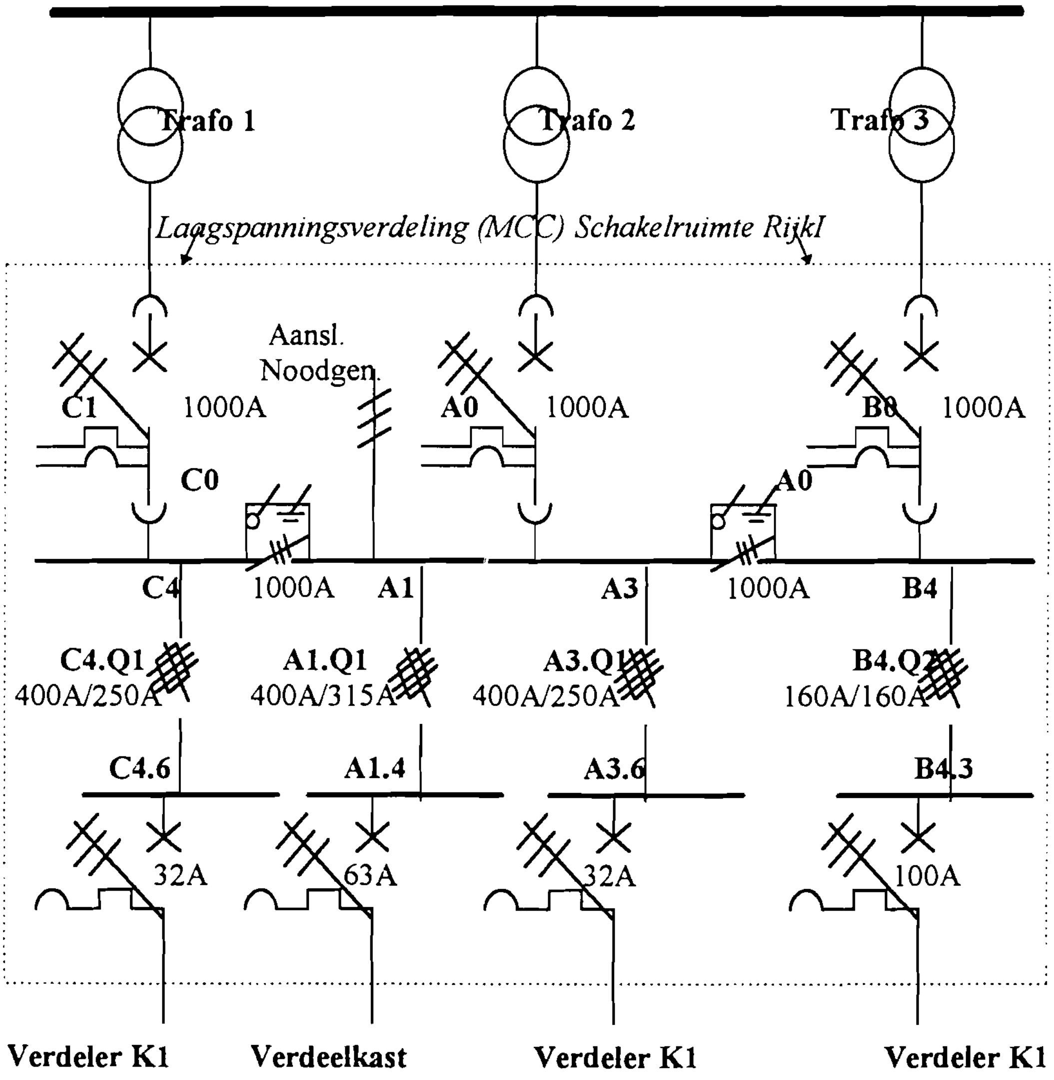
AFS piping system Oost to RijkI.



TANKOPSLAG OOST

Appendix F1

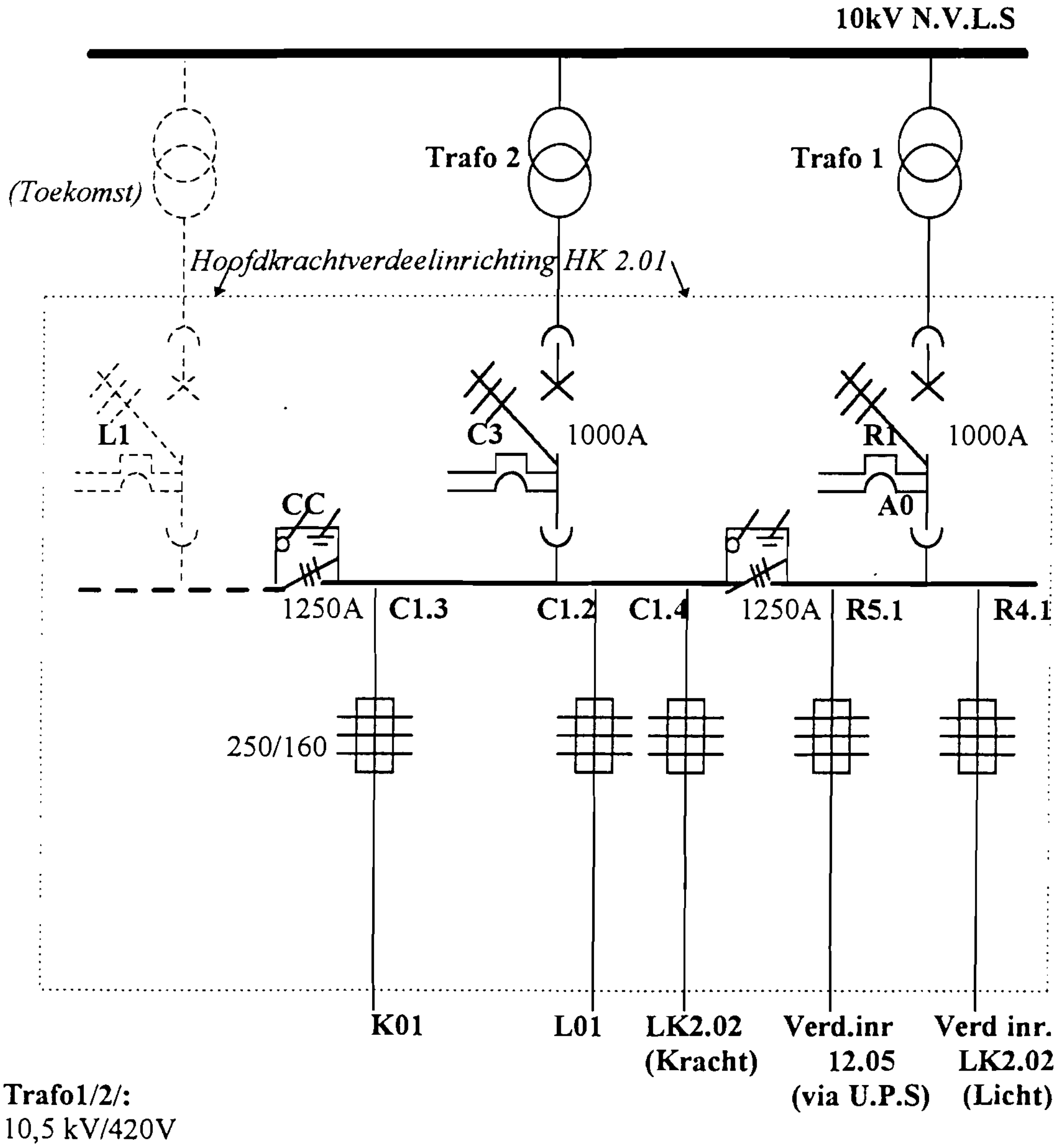
Single Line Diagram RijkI



Trafo1/2/3:
10,25 kV/0,4kV
630kVA Dyn5
In=909A
4%

Appendix F2

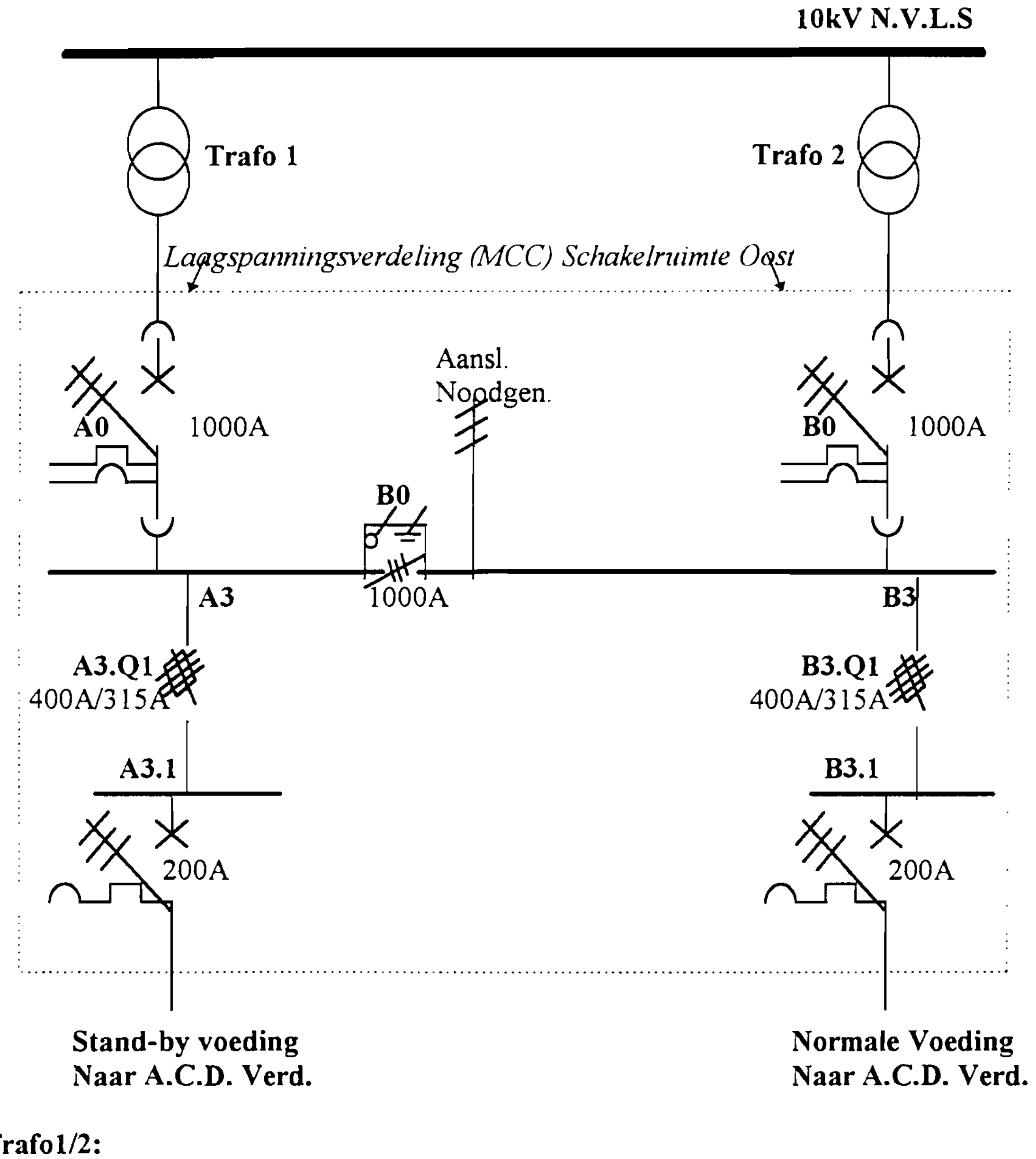
Single Line Diagram RijkII



Trafo1/2/:
10,5 kV/420V
630kVA Dyn11
In=909A
6%

Appendix F3

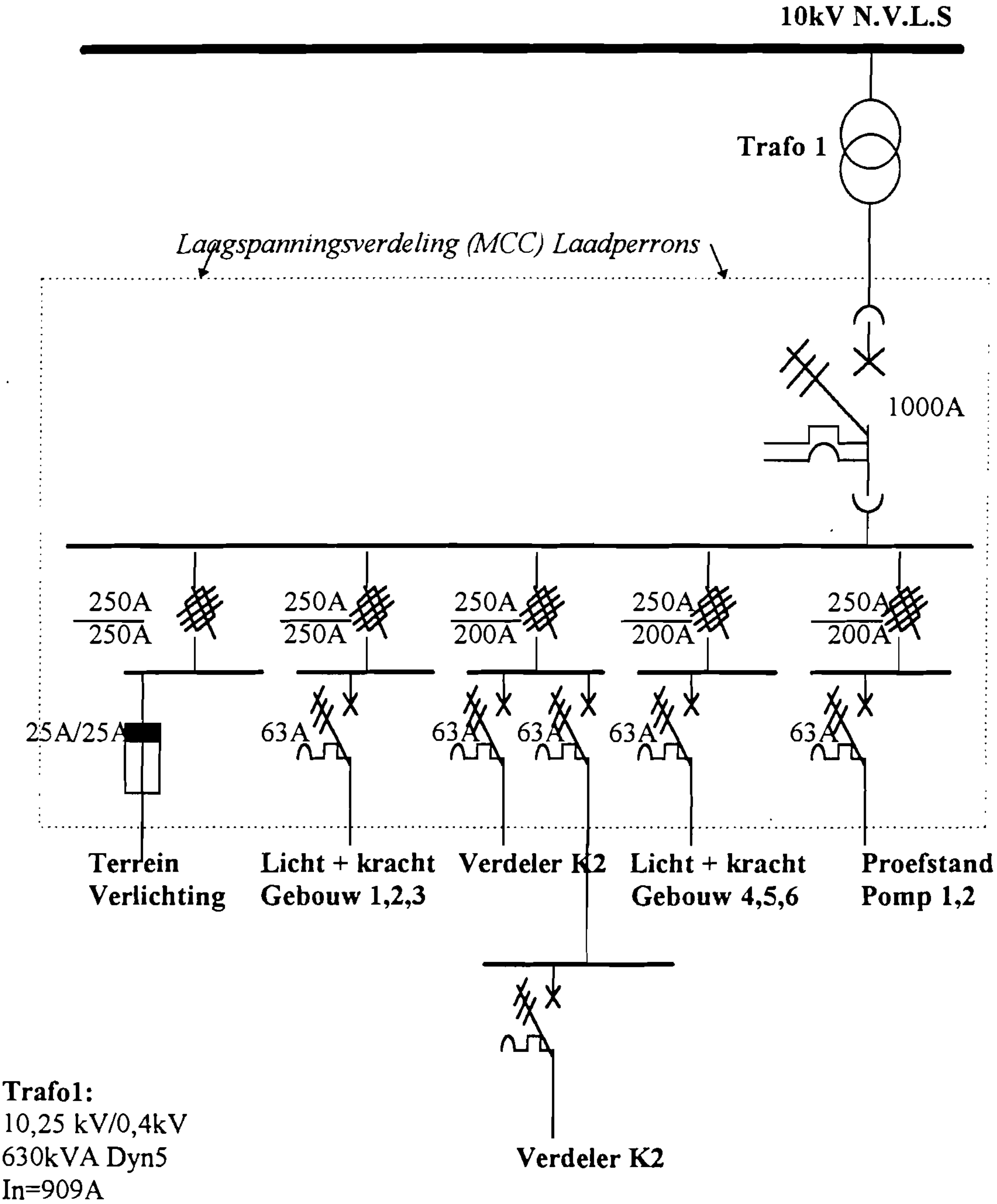
Single Line Diagram Oost



Trafo1/2:
10,25 kV/0,4kV
630kVA Dyn5
In=909A
4%

Appendix F4

Single Line Diagram Centrum



Trafo1:
10,25 kV/0,4kV
630kVA Dyn5
In=909A
4%

Appendix G1

Base comparison system.

- m is the amount of FSP's;
- Q_F is the flow in one FSP;
- P_{motor} is the absorbed power by the motor;
- T is the amount of hours in a week by a certain flowrate;
- E is the Energy usage;

Q_{totaal} [m³/h]	m	Q_F [m³/h]	P [kW]	P_{motor} [kW]	T [h]	E [kWh]
50	1	50	52.7	55.47	0.306	16.97
75	1	75	59.3	62.42	2.401	149.87
100	1	100	65.6	69.05	2.894	199.83
125	1	125	71.7	75.47	3.574	269.73
150	1	150	77.6	81.68	0.526	42.96
175	1	175	83.3	87.68	3.133	274.7
200	1	200	88.7	93.37	7.92	739.49
225	1	225	94	98.95	0.614	60.76
250	1	250	99	104.21	6.268	653.19
275	1	275	103.8	109.26	4.846	529.47
300	2	150	155.2	163.37	3.056	499.26
350	2	175	166.6	175.37	9.884	1733.36
400	2	200	177.4	186.74	4.798	895.98
450	2	225	188	197.89	10.526	2082.99
500	2	250	198	208.42	3.624	755.31
550	2	275	207.6	218.53	5.682	1241.69
600	3	200	266.1	280.11	3.637	1018.76
675	3	225	282	296.84	8.77	2603.29
750	3	250	297	312.63	4.068	1271.78
825	3	275	311.4	327.79	3.987	1306.9
900	4	225	376	395.79	5.763	2280.94
1000	4	250	396	416.84	4.628	1929.14
1100	4	275	415.2	437.05	3.994	1745.58
1250	5	250	495	521.05	4.992	2601.08
1375	5	275	519	546.32	3.244	1772.26
1500	6	250	594	625.26	1.993	1246.14
1650	6	275	622.8	655.58	1.362	892.9
1800	7	257.1	702.7	739.68	0.423	312.88
1900	7	271.4	721.8	759.79	0.232	176.27
2150	8	268.7	820.8	864	0.105	90.72

From this approximation: E_{totaal} = 29394.2 kWh a week.

Appendix G2

1 VSP and m FSP's.

- $Q_{F1} = Q_{Fm} = 288\ m^3/h$, mits $Q_V \geq 36\ m^3/h$
- $Q_V = Q_{totaal} - m \cdot Q_F$
- $P_{FSP}(288) = 106.2$ kW
- $P_{tot} = m \cdot P_{FSP} + P_{VSP}$ $\quad P_{motor} = P_{tot}/\eta$
- $H_{VSP}(Q_V) = H_{FSP}(Q_F) = 122.1$ meter

  $171 \cdot n_V^2 + 48.1 \cdot n_V \cdot Q_V - 3710 \cdot Q_V^2 = 122.1 \cdot 10^7$ if $Q_F = 288 m^3/h$

Q_{totaal} [m^3/h]	m	Q_F [m^3/h]	Q_V [m^3/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	1	**264**	36		2722	39.1	140.8	3.056	46.32
350	1	288	62		2648	41.8	148	9.884	193.52
400	1	288	112		2707	54.9	161.1	4.798	82.32
450	1	288	162		2754	67.5	173.3	10.526	162.88
500	1	288	212		2819	81.5	187.7	3.624	39.29
550	1	288	262		2901	97.2	203.4	5.682	25.12
600	2	**282**	36		2685	37.7	250.1	3.637	61.25
675	2	288	99		2698	51.8	264.2	8.77	164.32
750	2	288	174		2768	70.7	283.1	4.068	59.52
825	2	288	249		2878	92.9	305.3	3.987	25.6
900	3	288	36		2672	37.2	355.8	5.763	122.54
1000	3	288	136		2727	60.8	379.4	4.628	80.87
1100	3	288	236		2856	88.7	407.3	3.994	32.21
1250	4	288	98		2697	51.5	476.3	4.992	98.26
1375	4	288	223		2836	84.8	509.6	3.244	32.1
1500	5	288	60		2678	42.6	573.6	1.993	42.8
1650	5	288	210		2816	80.9	611.9	1.362	15.63
1800	6	288	72		2682	45.4	682.6	0.423	8.95
1900	6	288	172		2765	70.1	707.3	0.232	3.54
2150	7	288	134		2725	60.3	803.7	0.105	1.89

Total $E_{savings}$ [kWh] in one week: **2238 kWh**
Total $E_{savings}$ [kWh] in one year: **116700 kWh**

Savings in guilders:

Energy price savings: 116700 * 0.0680 = *f* 7936,-
Savings on fuel : jan t/m march: 0.0561 * $E_{savings\ jan\text{-}ma}$ +
april t/m sept: 0.0535 * $E_{savings\ apr\text{-}sept}$ +
oct t/m dec: 0.0546 * $E_{savings\ oct\text{-}dec}$ =
= 2,838 * ($E_{savings}$ in one week) = *f* 6351,-

Total: *f* 14287,- + savings on peak power + savings contractual power

Appendix G3

2 VSP's and m FSP's.

- $Q_{F1} = Q_{Fm} = 288\ m^3/h$, if $Q_V \geq 36\ m^3/h$
- $Q_V = 1/2(Q_{totaal} - m \cdot Q_F)$
- $P_{FSP}(288) = 106.2$ kW
- $P_{tot} = m \cdot P_{FSP} + 2 \cdot P_{VSP}$
- $H_{VSP}(Q_V) = H_{FSP}(Q_F) = 122.1$ meter

  $171 \cdot n_V^2 + 48.1 \cdot n_V \cdot Q_V - 3710 \cdot Q_V^2 = 122.1 \cdot 10^7$ if $Q_F = 288 m^3/h$

Q_{totaal} [m^3/h]	m	Q_F [m^3/h]	Q_V [m^3/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	1	288	156		2747	65.9	238	3.637	107.58
675	1	288	193.5		2793	76.1	258.4	8.77	217.86
750	1	288	231		2848	87.2	280.6	4.068	70.22
825	1	288	268.5		2913	99.4	305	3.987	26.86
900	2	288	162		2754	67.5	347.4	5.763	173.5
1000	2	288	212		2819	81.5	375.4	4.628	100.35
1100	2	288	262		2901	97.2	406.8	3.994	35.32
1250	3	288	193		2792	76	470.6	4.992	128.22
1375	3	288	255.5		2889	95	508.6	3.244	35.51
1500	4	288	174		2768	70.7	566.2	1.993	58.32
1650	4	288	249		2878	92.9	610.6	1.362	17.49
1800	5	288	180		2775	72.3	675.6	0.423	12.07
1900	5	288	230		2847	86.9	704.8	0.232	4.15
2150	6	288	211		2818	81.2	799.6	0.105	2.34

Total $E_{savings}$ [kWh] in one week: **4118 kWh**
Total $E_{savings}$ [kWh] in one year: **214738 kWh**

Savings in guilders:

Energy price savings: 214738 * 0.0680 = *f* 14602,-
Savings on fuel : 2,838 * ($E_{savings}$ in one week) = *f* 11687,-
Total: *f* 26289,- + savings on peak power + savings on contractual power

Appendix G4

3 VSP's and m FSP's.

$Q_{F1} = Q_{Fm} = 288\ m^3/h$, mits $Q_V \geq 36\ m^3/h$
$Q_V = 1/3(Q_{totaal} - m \cdot Q_F)$

$P_{tot} = m \cdot P_{FSP} + 3 \cdot P_{VSP}$

Q_{totaal} [m³/h]	m	Q_F [m³/h]	Q_V [m³/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	0	0	200	96.9	2529	60.30	180.9	3.637	326.23
675	0	0	225	97.4	2575	66.83	200.5	8.77	752.57
750	0	0	250	97.8	2625	73.89	221.7	4.068	322.59
825	0	0	275	98.2	2679	81.54	244.6	3.987	280.32
900	1	288	204		2808	79.2	343.8	5.763	195.34
1000	1	288	237.3		2859	89.2	373.8	4.628	108.15
1100	1	288	270.7		2919	100.3	407.1	3.994	34.05
1250	2	288	224.7		2838	85.2	468	4.992	141.88
1375	2	288	266.3		2909	98.6	508.2	3.244	36.88
1500	3	288	212		2819	81.5	563.1	1.993	64.82
1650	3	288	262		2901	97.2	610.2	1.362	18.06
1800	4	288	216		2825	82.6	672.6	0.423	13.4
1900	4	288	249.3		2879	93	703.8	0.232	4.4
2150	5	288	236.7		2858	89	798	0.105	2.52

Total $E_{savings}$ [kWh] in one week:	**5430 kWh**
Total $E_{savings}$ [kWh] in one year:	**283119 kWh**

Savings in guilders:

Energy price savings:	283119 * 0.0680 = *f* 19252,-
Savings on fuel :	2,838 * ($E_{savings}$ in one week) = *f* 15410,-

Total: *f* 34662,- + savings on peak power + savings contractual power

Appendix G5

4 VSP's and m FSP's.

- $Q_{F1} = Q_{Fm} = 288$ m^3/h, mits $Q_V \geq 36$ m^3/h
- $Q_V = 1/4(Q_{totaal} - m \cdot Q_F)$
- $P_{tot} = m \cdot P_{FSP} + 4 \cdot P_{VSP}$

Q_{totaal} [m^3/h]	m	Q_F [m^3/h]	Q_V [m^3/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	0	0	200	96.9	2529	60.30	180.9	3.637	326.23
675	0	0	225	97.4	2575	66.83	200.5	8.77	752.57
750	0	0	250	97.8	2625	73.89	221.7	4.068	322.59
825	0	0	275	98.2	2679	81.54	244.6	3.987	280.32
900	0	0	225	98.7	2589	67.78	271.1	5.763	636.19
1000	0	0	250	99.4	2642	75.13	300.5	4.628	465.12
1100	0	0	275	100.1	2700	83.12	249.4	3.994	347.72
1250	1	288	240.5		2864	90.2	467	4.992	147.13
1375	1	288	271.8		2919	100.5	508.2	3.244	36.88
1500	2	288	231		2848	87.2	561.2	1.993	68.81
1650	2	288	268.5		2913	99.4	610	1.362	18.35
1800	3	288	234		2853	88.1	671	0.423	14.11
1900	3	288	259		2896	96.2	703.4	0.232	4.49
2150	4	288	249.5		2879	93.1	797.2	0.105	2.61

Total $E_{savings}$ [kWh] in one week: **6552 kWh**
Total $E_{savings}$ [kWh] in one year: **341619 kWh**

Savings in guilders:

Energy price savings: 341619 * 0.0680 = *f* 23230,-
Savings on fuel : 2,838 * ($E_{savings}$ in one week) = *f* 18595,-
Total: *f* 41825,- + savings on peak power + savings contractual power

Appendix G6

5 VSP's and m FSP's.

- $Q_{F1} = Q_{Fm} = 288$ m^3/h, mits $Q_V \geq 36$ m^3/h
- $Q_V = 1/5(Q_{totaal} - m \cdot Q_F)$
- $P_{tot} = m \cdot P_{FSP} + 5 \cdot P_{VSP}$

Q_{totaal} [m^3/h]	m	Q_F [m^3/h]	Q_V [m^3/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	0	0	200	96.9	2529	60.30	180.9	3.637	326.23
675	0	0	225	97.4	2575	66.83	200.5	8.77	752.57
750	0	0	250	97.8	2625	73.89	221.7	4.068	322.59
825	0	0	275	98.2	2679	81.54	244.6	3.987	280.32
900	0	0	225	98.7	2589	67.78	271.1	5.763	636.19
1000	0	0	250	99.4	2642	75.13	300.5	4.628	465.12
1100	0	0	275	100.1	2700	83.12	249.4	3.994	347.72
1250	0	0	250	101.4	2664	76.70	383.5	4.992	585.99
1375	0	0	275	102.5	2725	85.10	425.5	3.244	319.36
1500	1	288	242.4		2867	90.8	560.2	1.993	70.91
1650	1	288	272.4		2920	100.7	609.7	1.362	18.78
1800	2	288	244.8		2871	91.5	669.9	0.423	14.6
1900	2	288	264.8		2906	98.1	702.9	0.232	4.62
2150	3	288	257.2		2892	95.5	796.1	0.105	2.73

Total $E_{savings}$ [kWh] in one week: **7276 kWh**
Total $E_{savings}$ [kWh] in one year: **379402 kWh**

Savings in guilders:

Energy price savings: 379402 * 0.0680 = *f* 25800,-
Savings on fuel : 2,838 * ($E_{savings}$ in one week) = *f* 20649,-
Total: *f* 46449,- + savings on peak power + savings contractual power

Appendix G7

6 VSP's and m FSP's.

- $Q_{F1} = Q_{Fm} = 288\ m^3/h$, mits $Q_V \geq 36\ m^3/h$
- $Q_V = 1/6(Q_{totaal} - m \cdot Q_F)$
- $P_{tot} = m \cdot P_{FSP} + 6 \cdot P_{VSP}$

Q_{totaal} [m³/h]	m	Q_F [m³/h]	Q_V [m³/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	0	0	200	96.9	2529	60.30	180.9	3.637	326.23
675	0	0	225	97.4	2575	66.83	200.5	8.77	752.57
750	0	0	250	97.8	2625	73.89	221.7	4.068	322.59
825	0	0	275	98.2	2679	81.54	244.6	3.987	280.32
900	0	0	225	98.7	2589	67.78	271.1	5.763	636.19
1000	0	0	250	99.4	2642	75.13	300.5	4.628	465.12
1100	0	0	275	100.1	2700	83.12	249.4	3.994	347.72
1250	0	0	250	101.4	2664	76.70	383.5	4.992	585.99
1375	0	0	275	102.5	2725	85.10	425.5	3.244	319.36
1500	0	0	250	103.7	2690	78.55	471.3	1.993	257.39
1650	0	0	275	105.4	2755	87.47	524.8	1.362	140.47
1800	1	288	252		2883	93.8	669	0.423	15.01
1900	1	288	268.7		2913	99.4	702.6	0.232	4.69
2150	2	288	262.3		2902	97.3	796.2	0.105	2.72

Total $E_{savings}$ [kWh] in one week: **7585 kWh**
Total $E_{savings}$ [kWh] in one year: **395504 kWh**

Savings in guilders:

Energy price savings: 395504 * 0.0680 = *f* 26894,-
Savings on fuel : 2,838 * ($E_{savings}$ in one week) = *f* 21526,-
Total: *f* 48420,- + savings on peak power + savings contractual power

Appendix G8

7 VSP's and m FSP's.

- $Q_{F1} = Q_{Fm} = 288$ m^3/h, mits $Q_V \geq 36$ m^3/h
- $Q_V = 1/7(Q_{totaal} - m \cdot Q_F)$
- $P_{tot} = m \cdot P_{FSP} + 7 \cdot P_{VSP}$

Q_{totaal} [m^3/h]	m	Q_F [m^3/h]	Q_V [m^3/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	0	0	200	96.9	2529	60.30	180.9	3.637	326.23
675	0	0	225	97.4	2575	66.83	200.5	8.77	752.57
750	0	0	250	97.8	2625	73.89	221.7	4.068	322.59
825	0	0	275	98.2	2679	81.54	244.6	3.987	280.32
900	0	0	225	98.7	2589	67.78	271.1	5.763	636.19
1000	0	0	250	99.4	2642	75.13	300.5	4.628	465.12
1100	0	0	275	100.1	2700	83.12	249.4	3.994	347.72
1250	0	0	250	101.4	2664	76.70	383.5	4.992	585.99
1375	0	0	275	102.5	2725	85.10	425.5	3.244	319.36
1500	0	0	250	103.7	2690	78.55	471.3	1.993	257.39
1650	0	0	275	105.4	2755	87.47	524.8	1.362	140.47
1800	0	0	257	107.1	2739	83.35	583.5	0.423	53.11
1900	0	0	271	108.3	2778	88.76	621.3	0.232	24.55
2150	1	288	266		2908	98.5	795.7	0.105	2.77

Total $E_{savings}$ [kWh] in one week:	**7643 kWh**
Total $E_{savings}$ [kWh] in one year:	**398520 kWh**

Savings in guilders:

Energy price savings:	398520 * 0.0680 = *f* 27100,-
Savings on fuel :	2,838 * ($E_{savings}$ in one week) = *f* 21691,-

Total: *f* 48791,- + savings on peak power + savings contractual power

Appendix G9

8 VSP's (no FSP's).

- $Q_V = 1/7(Q_{totaal} - m \cdot Q_F)$
- $P_{tot} = 8 \cdot P_{VSP}$
- If the flowrate in a month is bigger than 8*288 = 2304 m^3/h, the savings (peak power) giving below are not valid.

Q_{totaal} [m^3/h]	m	Q_F [m^3/h]	Q_V [m^3/h]	H_{vtot} [m]	n_V [rpm]	P_V [kW]	P_{tot} [kW]	T [h]	$E_{savings}$ [kWh]
50	0	0	50	95.5	2368	28.97	28.97	0.306	7.64
75	0	0	75	95.5	2379	33.51	33.51	2.401	65.19
100	0	0	100	95.6	2395	38.18	38.18	2.894	83.52
125	0	0	125	95.6	2418	43.05	43.05	3.574	107.77
150	0	0	150	95.6	2445	48.18	48.18	0.526	16.29
175	0	0	175	95.7	2477	53.62	53.62	3.133	97.85
200	0	0	200	95.7	2515	59.44	59.44	7.92	243.96
225	0	0	225	95.8	2557	65.68	65.68	0.614	18.31
250	0	0	250	95.8	2603	72.40	72.40	6.268	175.51
275	0	0	275	95.9	2654	79.65	79.65	4.846	123.17
300	0	0	150	95.9	2448	48.35	96.7	3.056	188.21
350	0	0	175	96.1	2482	53.86	107.7	9.884	612.52
400	0	0	200	96.2	2520	59.77	119.5	4.798	292.24
450	0	0	225	96.4	2564	66.12	132.2	10.526	617.78
500	0	0	250	96.6	2612	72.98	145	3.624	198.55
550	0	0	275	96.8	2664	80.38	160.8	5.682	280.17
600	0	0	200	96.9	2529	60.30	180.9	3.637	326.23
675	0	0	225	97.4	2575	66.83	200.5	8.77	752.57
750	0	0	250	97.8	2625	73.89	221.7	4.068	322.59
825	0	0	275	98.2	2679	81.54	244.6	3.987	280.32
900	0	0	225	98.7	2589	67.78	271.1	5.763	636.19
1000	0	0	250	99.4	2642	75.13	300.5	4.628	465.12
1100	0	0	275	100.1	2700	83.12	249.4	3.994	347.72
1250	0	0	250	101.4	2664	76.70	383.5	4.992	585.99
1375	0	0	275	102.5	2725	85.10	425.5	3.244	319.36
1500	0	0	250	103.7	2690	78.55	471.3	1.993	257.39
1650	0	0	275	105.4	2755	87.47	524.8	1.362	140.47
1800	0	0	257	107.1	2739	83.35	583.5	0.423	53.11
1900	0	0	271	108.3	2778	88.76	621.3	0.232	24.55
2150	0	0	269	111.5	2806	90.52	724.2	0.105	10.68

Total $E_{savings}$ [kWh] in one week: **7653 kWh**
Total $E_{savings}$ [kWh] in one year: **399050 kWh**

Savings in guilders:

Energy price savings: 399050 * 0.0680 = *f* 27135,-
Savings on fuel : 2,838 * ($E_{savings}$ in one week) = *f* 21719,-
Total: *f* 48854,- + savings on peak power + savings contractual power

Appendix H1

Total investment Costs 1 VSP

• Ex(d) Motor + geforceerde koeling:	*f* 35.000,-	(vlgs. Shell Wedci boekje)
• Bekabeling:	*f* 26.000,-	(vlgs Prijsbudgetopg. R&H)
• Installatie geforceerde koeling:	*f* 1.500,-	(vlgs Prijsbudgetopg. R&H)
• Kast + VFD:	*f* 60.000,-	(vlgs Prijsbudgetopg. R&H)
• Voedingsveld:	*f* 5.000,-	(vlgs Prijsbudgetopg. R&H)
• Bouwkundige kosten:	*f* 21.000,-	(vlgs Prijsbudgetopg. R&H)
• Besturing pomp:	*f* 1.000,-	(vlgs Prijsbudgetopg. R&H)
• Computerprogramma:	*f* 15.000,-	
Totaal:	*f* 164.500,-	
• 10 % onvoorzien:	*f* 16.500,-	
Total investment:	***f* 181.000,-**	

Appendix H2

Total investment Costs 2 VSP's

• Ex(d) Motor + geforceerde koeling:	*f* 70.000,-	(vlgs. Shell Wedci boekje)
• Bekabeling:	*f* 46.250,-	(vlgs Prijsbudgetopg. R&H)
• Installatie geforceerde koeling:	*f* 3.000,-	(vlgs Prijsbudgetopg. R&H)
• Kast + VFD:	*f* 100.000,-	(vlgs Prijsbudgetopg. R&H)
• Voedingsveld:	*f* 10.000,-	(vlgs Prijsbudgetopg. R&H)
• Bouwkundige kosten:	*f* 21.000,-	(vlgs Prijsbudgetopg. R&H)
• Besturing pomp:	*f* 2.000,-	(vlgs Prijsbudgetopg. R&H)
• Computerprogramma:	*f* 15.000,-	
Totaal:	*f* 267.250,-	
• 10 % onvoorzien:	*f* 26.750,-	
Total investment:	***f* 294.000,-**	

Appendix H3

Total investment Costs 3 VSP's

• Ex(d) Motor + geforceerde koeling:	*f* 105.000,-	(vlgs. Shell Wedci boekje)
• Bekabeling:	*f* 63.750,-	(vlgs Prijsbudgetopg. R&H)
• Installatie geforceerde koeling:	*f* 4.500,-	(vlgs Prijsbudgetopg. R&H)
• Kast + VFD:	*f* 160.000,-	(vlgs Prijsbudgetopg. R&H)
• Voedingsveld:	*f* 15.000,-	(vlgs Prijsbudgetopg. R&H)
• Bouwkundige kosten:	*f* 21.000,-	(vlgs Prijsbudgetopg. R&H)
• Besturing pomp:	*f* 3.000,-	(vlgs Prijsbudgetopg. R&H)
• Computerprogramma:	*f* 15.000,-	
Totaal:	*f* 286.250,-	
• 10 % onvoorzien:	*f* 38.750,-	
Total investment:	***f* 425.000,-**	

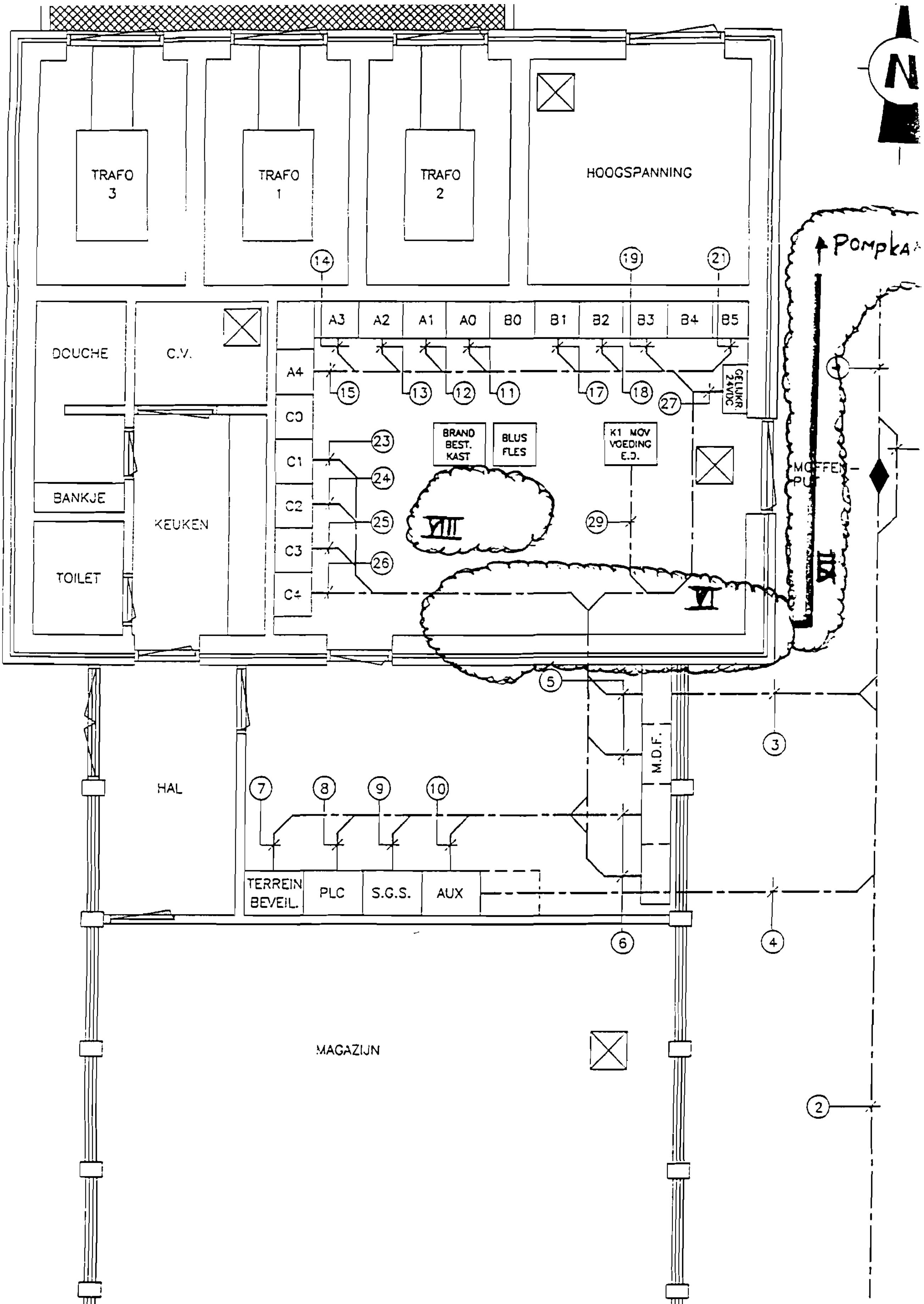
Appendix H4

Total investment Costs 4 VSP's

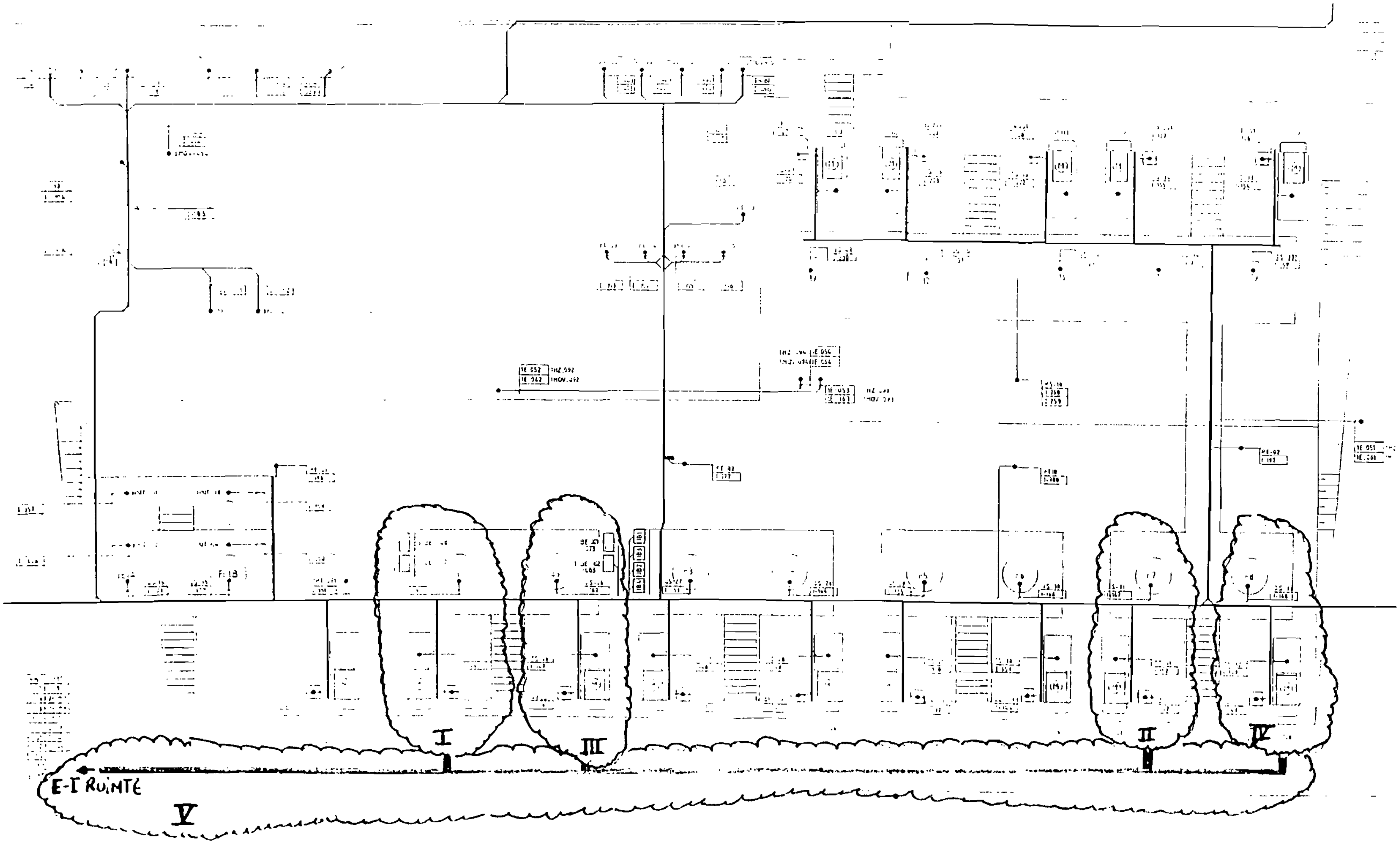
• Ex(d) Motor + geforceerde koeling:	*f* 140.000,-	(vlgs. Shell Wedci boekje)
• Bekabeling:	*f* 81.250,-	(vlgs Prijsbudgetopg. R&H)
• Installatie geforceerde koeling:	*f* 6.000,-	(vlgs Prijsbudgetopg. R&H)
• Kast + VFD:	*f* 200.000,-	(vlgs Prijsbudgetopg. R&H)
• Voedingsveld:	*f* 20.000,-	(vlgs Prijsbudgetopg. R&H)
• Bouwkundige kosten:	*f* 21.000,-	(vlgs Prijsbudgetopg. R&H)
• Besturing pomp:	*f* 4.000,-	(vlgs Prijsbudgetopg. R&H)
• Computerprogramma:	*f* 15.000,-	
Totaal:	*f* 487.250,-	
• 10 % onvoorzien:	*f* 48.750,-	
Total investment:	***f* 536.000,-**	

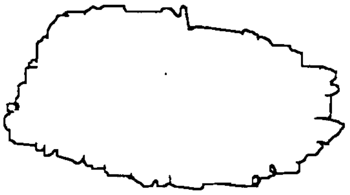
Appendix I

Drawings of the new installations


N
TRAFO 3
TRAFO 1
TRAFO 2
HOOGSPANNING
Pompka
DOUCHE
C.V.
BANKJE
TOILET
KEUKEN
A3 A2 A1 A0 B0 B1 B2 B3 B4 B5
A4
C0
C1
C2
C3
C4
BRAND BEST. KAST
BLUS FLES
K1 MOV VOEDING E.D.
GELIJKR. 24VDC
MOFFEN PUT
VIII
VI
IX
M.D.F.
HAL
TERREIN BEVEIL.
PLC
S.G.S.
AUX
MAGAZIJN
E&I

Pump department



Nog te installeren: 

1 VSP:

I:
Pomp 101

- Installatie van nieuwe Ex(d) motor
- Installatie van de geforceerde koeling op de motor
- Aanpassen van de pompbesturing

V + VII:
Aanleggen van kabelgoot + voedingskabel van VFD naar motor pomp 101

VI:

- Installatie van kast + VFD
- Aanleggen van voedingsveld

VII:
Installatie van Airco om de VFD te koelen

2 VSP's:

I + II:
Pomp 101 en pomp 107

- Installatie van nieuwe Ex(d) motor
- Installatie van de geforceerde koeling op de motor
- Aanpassen van de pompbesturing

V + VII:
Aanleggen van kabelgoot + 2 voedingskabels van VFD naar motor pomp 101 en naar motor pomp 107

VI:

- Installatie van 1 kast + 2 VFD's
- Aanleggen van 2 voedingsvelden

VII:
Installatie van Airco om de VFD te koelen

3 VSP's:

I + II + III:
Pomp 101, pomp 102 en pomp 107
- Installatie van nieuwe Ex(d) motor
- Installatie van de geforceerde koeling op de motor
- Aanpassen van de pompbesturing

V + VII:
Aanleggen van kabelgoot + 3 voedingskabels van VFD naar motor pomp 101, 102 en 107

VI:
- Installatie van 2 kast + 3 VFD's
- Aanleggen van 3 voedingsvelden

VII:
Installatie van Airco om de VFD te koelen

4 VSP's:

I + II + III + IV:
Pomp 101, 102, 107 en pomp 108
- Installatie van nieuwe Ex(d) motor
- Installatie van de geforceerde koeling op de motor
- Aanpassen van de pompbesturing

V + VII:
Aanleggen van kabelgoot + 4 voedingskabels van VFD naar motor pomp 101, 102, 107 en 108

VI:
- Installatie van 2 kast + 4 VFD's
- Aanleggen van 4 voedingsvelden

VII:
Installatie van Airco om de VFD te koelen

Additionele informatie:
- Pomp 101 t/m 106 pompen uit een tank op RijkI
- Pomp 107 t/m 108 pompen uit een tank op RijkII
- Pomp 101 en 102 hebben de oudste motors

Appendix J

Calculation of the payback period and the Discounted cash flow rate of Return

Summary of Appendix J:

Situation	No Subsidies		Subsidies	
	STP [year]	DRR [%]	STP [year]	DRR [%]
1 VSP	7,2	8,9	6,4	**12,5**
2 VSP's	7,1	9,4	6,4	**12,9**
3 VSP's	7,7	7,1	7	9,8
4 VSP's	8	6	7,2	8,2

One VSP and no Subsidies

Investment costs:	*f* 181.000,-
Annual power cost increase:	$(1,1)^n$, where n equals the year.
Corporate tax rate:	35%
Tax life of equipment:	10 year
Depreciation:	Straight line
Savings first year:	*f* 14.300,- * 1,1 + *f* 4.500,- = *f* 20.000,-

Year	**Savings [*f* 1000,-]**	**Depreciation [*f* 1000,-]**	**Taxable Income [*f* 1000,-]**	**Income Tax [*f* 1000,-]**	**After-Tax Cash flow [*f* 1000,-]**	**Accum'D Cash flow [*f* 1000,-]**
0	-181	0	0	0	-181	-181
1	20	18,1	1,9	0,7	19,3	-161,7
2	27,5	18,1	9,4	3,3	24,2	-137,5
3	26,4	18,1	8,3	2,9	23,5	-114
4	28,3	18,1	10,2	3,6	24,7	-89,2
5	30,4	18,1	12,3	4,3	26,1	-63,1
6	32,7	18,1	14,6	5,1	27,6	-35,5
7	35,3	18,1	17,2	6	29,3	-6,3
8	38,1	18,1	20	7	31,1	**24,8**
9	41,1	18,1	23	8,1	33,1	**57,9**
10	44,5	18,1	26,4	9,2	35,3	**93,1**

Discounted Rate of Return = DRR

$$\frac{19{,}3}{1+DRR}+\frac{24{,}2}{(1+DRR)^2}+\frac{23{,}5}{(1+DRR)^3}+\ldots+\frac{33{,}1}{(1+DRR)^9}+\frac{35{,}3}{(1+DRR)^{10}}=18$$

So that: $\mathbf{DRR_{10} = 8{,}9\%}$

The payback period is 7,2 year

One VSP and Subsidies

Investment costs: *f* 181.000,-
Annual power cost increase: $(1,1)^n$, where n equals the year.
Corporate tax rate: 35 %
Tax life of equipment: 10 year
Savings first year: *f* 14.300,- * 1,1 + *f* 4.500,- = *f* 20.000,-
Subsidies:

- EIA: 47,5 %
- Investment tax credit: 16 %
- VAMIL-depreciation

Year	Savings [*f* 1000,-]	Depreciation [*f* 1000,-]	Taxable Income [*f* 1000,-]	Income Tax [*f* 1000,-]	After-Tax Cash flow [*f* 1000,-]	Accum'D Cash flow [*f* 1000,-]
0	-181	0	0	0	-181	-181
1	20	20	0	0	20	-161
2	27,5	27,5	0	0	27,5	-133,5
3	26,4	26,4	0	0	26,4	-107,1
4	28,3	28,3	0	0	28,3	-78,8
5	30,4	30,4	0	0	30,4	-48,4
6	32,7	32,7	0	0	32,7	-15,7
7	35,3	35,3	0	0	35,3	**19,6**
8	38,1	38,1	0	0	38,1	**57,7**
9	41,1	41,1	0	0	41,1	**98,8**
10	44,5	16,2	28,3	9,9	34,6	**133,4**

Discounted Rate of Return = DRR

$$\frac{20}{1+DRR}+\frac{27{,}5}{(1+DRR)^2}+\frac{26{,}4}{(1+DRR)^3}+\ldots+\frac{38{,}1}{(1+DRR)^9}+\frac{41{,}1}{(1+DRR)^{10}}=18$$

So that: $\mathbf{DRR_{10} = 12{,}5\%}$

The payback period is 6,4 year

Two VSP's and no Subsidies

Investment costs:	*f* 294.000,-
Annual power cost increase:	$(1,1)^n$, where n equals the year.
Corporate tax rate:	35%
Tax life of equipment:	10 year
Depreciation:	Straight line
Savings first year:	*f* 26.300,- * 1,1 + *f* 6.000,- = *f* 34.900,-

Year	**Savings [*f* 1000,-]**	**Depreciation [*f* 1000,-]**	**Taxable Income [*f* 1000,-]**	**Income Tax [*f* 1000,-]**	**After-Tax Cash flow [*f* 1000,-]**	**Accum'D Cash flow [*f* 1000,-]**
0	-294	0	0	0	-294	-294
1	34,9	29,4	5,5	1,9	33	-261
2	42,1	29,4	12,7	4,4	37,7	-223,4
3	43,2	29,4	13,8	4,8	38,4	-185
4	46,7	29,4	17,3	6,1	40,6	-144,4
5	50,6	29,4	21,2	7,4	43,2	-101,2
6	54,8	29,4	25,4	8,9	45,9	-55,3
7	59,5	29,4	30,1	10,5	49	-6,3
8	64,6	29,4	35,2	12,3	52,3	**46**
9	70,2	29,4	40,8	14,3	55,9	**101,9**
10	76,4	29,4	47	16,5	60	**161,9**

Discounted Rate of Return = DRR

$$\frac{33}{1+DRR}+\frac{37,7}{(1+DRR)^2}+\frac{38,4}{(1+DRR)^3}+\ldots+\frac{55,9}{(1+DRR)^9}+\frac{60}{(1+DRR)^{10}}=294$$

So that: $\mathbf{DRR_{10} = 9,4\%}$

The payback period is 7,1 year

Two VSP's and Subsidies

Investment costs:	*f* 294.000,-
Annual power cost increase:	$(1,1)^n$, where n equals the year.
Corporate tax rate:	35 %
Tax life of equipment:	10 year
Savings first year:	*f* 26.300,- * 1,1 + *f* 6.000,- = *f* 34.900,-
Subsidies:	
• EIA:	46 %
• Investment tax credit:	13 %
• VAMIL-depreciation	

Year	Savings [*f* 1000,-]	Depreciation [*f* 1000,-]	Taxable Income [*f* 1000,-]	Income Tax [*f* 1000,-]	After-Tax Cash flow [*f* 1000,-]	Accum'D Cash flow [*f* 1000,-]
0	-294	0	0	0	-294	-294
1	34,9	34,9	0	0	34,9	-259,1
2	42,1	42,1	0	0	42,1	-217
3	43,2	43,2	0	0	43,2	-173,8
4	46,7	46,7	0	0	46,7	-127,1
5	50,6	50,6	0	0	50,6	-76,5
6	54,8	54,8	0	0	54,8	-21,7
7	59,5	59,5	0	0	59,5	**37,8**
8	64,6	64,6	0	0	64,6	**102,4**
9	70,2	70,2	0	0	70,2	**172,6**
10	76,4	0,4	76	26,6	49,8	**222,4**

Discounted Rate of Return = DRR

$$\frac{34,9}{1+DRR}+\frac{42,1}{(1+DRR)^2}+\frac{43,2}{(1+DRR)^3}+\ldots+\frac{70,2}{(1+DRR)^9}+\frac{49,8}{(1+DRR)^{10}}=294$$

So that: $\mathbf{DRR_{10} = 12,9\%}$

The payback period is 6,4 year

Three VSP's and no Subsidies

Investment costs:	*f* 425.000,-
Annual power cost increase:	$(1,1)^n$, where n equals the year.
Corporate tax rate:	35%
Tax life of equipment:	10 year
Depreciation:	Straight line
Savings first year:	*f* 34.700,- * 1,1 + *f* 6.700,- = *f* 44.900,-

Year	**Savings [*f* 1000,-]**	**Depreciation [*f* 1000,-]**	**Taxable Income [*f* 1000,-]**	**Income Tax [*f* 1000,-]**	**After-Tax Cash flow [*f* 1000,-]**	**Accum'D Cash flow [*f* 1000,-]**
0	-425	0	0	0	-425	-425
1	44,9	42,5	2,4	0,8	44,1	-380,9
2	52,3	42,5	9,8	3,4	48,9	-332,1
3	54,7	42,5	12,2	4,3	50,4	-281,6
4	59,3	42,5	16,8	5,9	53,4	-228,2
5	64,4	42,5	21,9	7,7	56,7	-171,5
6	70	42,5	27,5	9,6	60,4	-111,1
7	76,1	42,5	33,6	11,8	64,3	-46,8
8	82,9	42,5	40,4	14,1	68,8	**22**
9	90,3	42,5	47,8	16,7	73,6	**95,6**
10	98,5	42,5	56	19,6	78,9	**174,5**

Discounted Rate of Return = DRR

$$\frac{44,1}{1+DRR}+\frac{48,9}{(1+DRR)^2}+\frac{50,4}{(1+DRR)^3}+\ldots+\frac{73,6}{(1+DRR)^9}+\frac{78,9}{(1+DRR)^{10}}=42$$

So that: $\mathbf{DRR_{10} = 7,1\%}$

The payback period is 7,7 year

Three VSP's and Subsidies

Investment costs:	*f* 425.000,-
Annual power cost increase:	$(1,1)^n$, where n equals the year.
Corporate tax rate:	35 %
Tax life of equipment:	10 year
Savings first year:	*f* 34.700,- * 1,1 + *f* 6.700,- = *f* 44.900,-
Subsidies:	
• EIA:	41,5 %
• Investment tax credit:	5 %
• VAMIL-depreciation	

Year	Savings [*f* 1000,-]	Depreciation [*f* 1000,-]	Taxable Income [*f* 1000,-]	Income Tax [*f* 1000,-]	After-Tax Cash flow [*f* 1000,-]	Accum'D Cash flow [*f* 1000,-]
0	-425	0	0	0	-425	-425
1	44,9	44,9	0	0	44,9	-380,1
2	52,3	52,3	0	0	52,3	-327,8
3	54,7	54,7	0	0	54,7	-273,1
4	59,3	59,3	0	0	59,3	-213,8
5	64,4	64,4	0	0	64,4	-149,4
6	70	70	0	0	70	-79,4
7	76,1	76,1	0	0	76,1	-3,3
8	82,9	82,9	0	0	82,9	**79,6**
9	90,3	90,3	0	0	90,3	**169,9**
10	98,5	28,1	70,4	26,6	73,9	**243,8**

Discounted Rate of Return = DRR

$$\frac{44{,}9}{1+DRR}+\frac{52{,}3}{(1+DRR)^2}+\frac{54{,}7}{(1+DRR)^3}+\ldots+\frac{82{,}9}{(1+DRR)^9}+\frac{90{,}3}{(1+DRR)^{10}}=42$$

So that: **DRR_{10} = 9,8%**

The payback period is 7 year

Four VSP's and no Subsidies

Investment costs:	*f* 536.000,-
Annual power cost increase:	$(1,1)^n$, where n equals the year.
Corporate tax rate:	35%
Tax life of equipment:	10 year
Depreciation:	Straight line
Savings first year:	*f* 41.800,- * 1,1 + *f* 7.100,- = *f* 53.100,-

Year	**Savings [*f* 1000,-]**	**Depreciation [*f* 1000,-]**	**Taxable Income [*f* 1000,-]**	**Income Tax [*f* 1000,-]**	**After-Tax Cash flow [*f* 1000,-]**	**Accum'D Cash flow [*f* 1000,-]**
0	-536	0	0	0	-536	-536
1	53,1	53,6	-0,5	0	53,1	-482,7
2	60,9	53,6	7,3	2,6	58,3	-424,4
3	64,3	53,6	10,7	3,7	60,6	-363,8
4	69,2	53,6	15,6	5,5	63,7	-300,1
5	76	53,6	22,4	7,8	68,2	-231,9
6	82,7	53,6	29,1	10,2	72,5	-159,4
7	90,2	53,6	36,6	12,8	77,4	-82
8	98,3	53,6	44,7	15,6	82,7	**0,6**
9	107,3	53,6	53,7	18,8	88,5	**89,1**
10	117,1	53,6	63,5	22,2	94,8	**184**

Discounted Rate of Return = DRR

$$\frac{53,1}{1+DRR}+\frac{58,3}{(1+DRR)^2}+\frac{60,6}{(1+DRR)^3}+...+\frac{88,5}{(1+DRR)^9}+\frac{94,9}{(1+DRR)^{10}}=536$$

So that: $\mathbf{DRR_{10} = 6,0\%}$

The payback period is 8 year

Four VSP's and Subsidies

Investment costs: *f* 536.000,-
Annual power cost increase: $(1,1)^n$, where n equals the year.
Corporate tax rate: 35 %
Tax life of equipment: 10 year
Savings first year: *f* 41.800,- * 1,1 + *f* 7.100,- = *f* 53.100,-
Subsidies:
- EIA: 40 %
- Investment tax credit: 0 %
- VAMIL-depreciation

Year	Savings [*f* 1000,-]	Depreciation [*f* 1000,-]	Taxable Income [*f* 1000,-]	Income Tax [*f* 1000,-]	After-Tax Cash flow [*f* 1000,-]	Accum'D Cash flow [*f* 1000,-]
0	-536	0	0	0	-536	-536
1	53,1	53,1	0	0	53,1	-482,9
2	60,9	60,9	0	0	60,9	-422
3	64,3	64,3	0	0	64,3	-357,7
4	69,2	69,2	0	0	69,2	-288,5
5	76	76	0	0	76	-212,5
6	82,7	82,7	0	0	82,7	-129,8
7	90,2	90,2	0	0	90,2	-39,6
8	98,3	98,3	0	0	98,3	**58,7**
9	107,3	107,3	0	0	107,3	**166**
10	117,1	48	69,1	24,2	92,9	**258,9**

Discounted Rate of Return = DRR

$$\frac{53{,}1}{1+DRR}+\frac{60{,}9}{(1+DRR)^2}+\frac{64{,}3}{(1+DRR)^3}+\ldots+\frac{107{,}3}{(1+DRR)^9}+\frac{92{,}9}{(1+DRR)^{10}}=536$$

So that: $\mathbf{DRR_{10} = 8{,}2\%}$

The payback period is 7,4 year