LIITE I

Labtium/Aalto-yliopisto
Mustalipeän ei-Newtonilaisuus ja pisaroituminen
– projektin tilanne 19.3.2014

Mustalipeän ei-Newtonilaisuus

AALTO&Labtium

Tehtävät

- Tehdään näytteenottolaite
- Otetaan näyte estäen höyryn karkaaminen
- Mitataan alustava viskositeetti tehtaalla
- Mitataan viskositeetti laboratoriossa
- Havu-, lehtipuu, seka- ja eucalyptuslipeä
- Mietitään tulosten merkitystä pisaroihin ym.

Näytteenotin

Näytteenottimen kiinnitys

Viskositeetin mittaus tehtaalla

Hagen-Poeiseuille

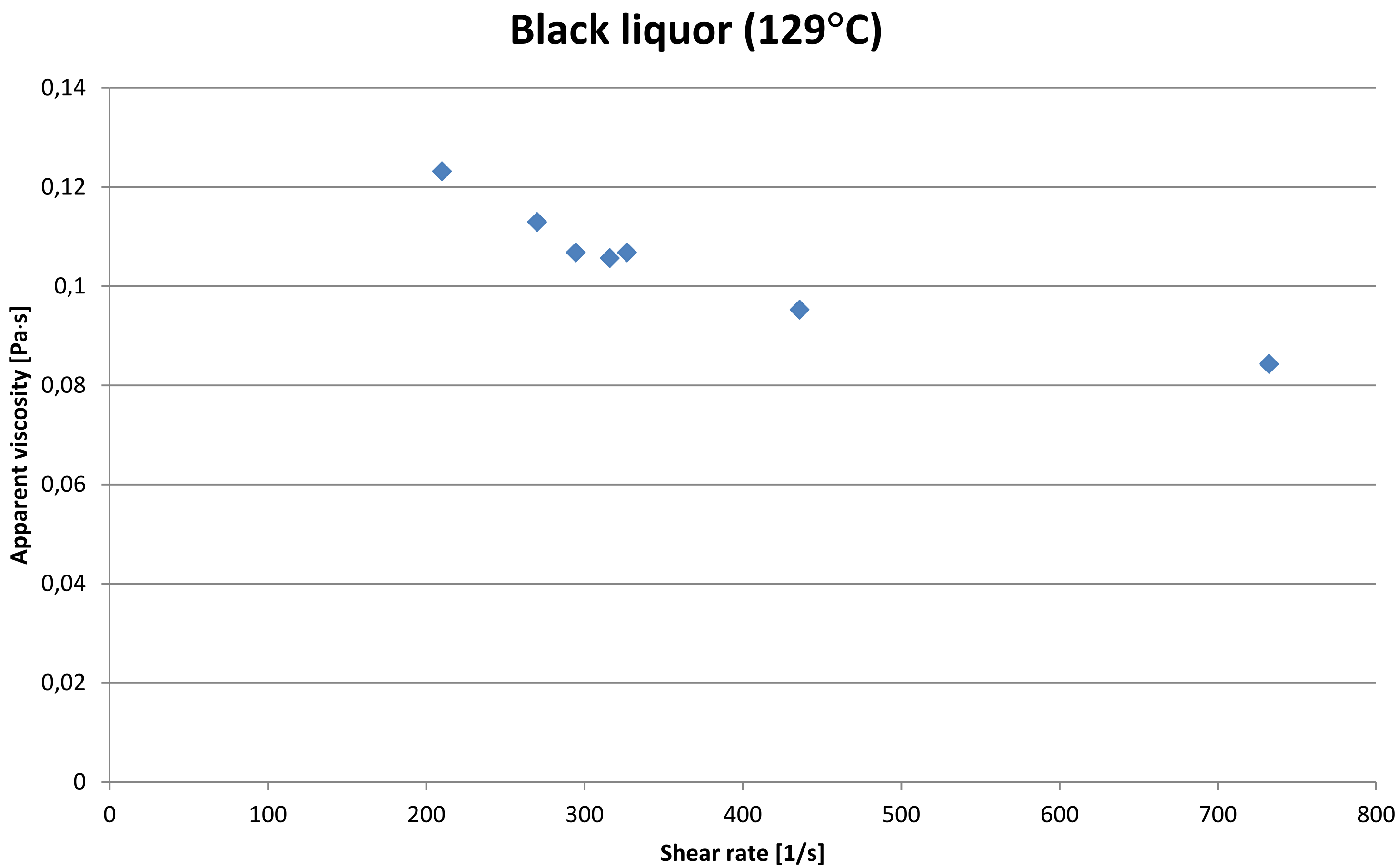
Hagen-Poiseuillen yhtälö on muotoa:

$$\Delta P = \frac{128\eta l q_v}{\pi d^4}$$

missä:

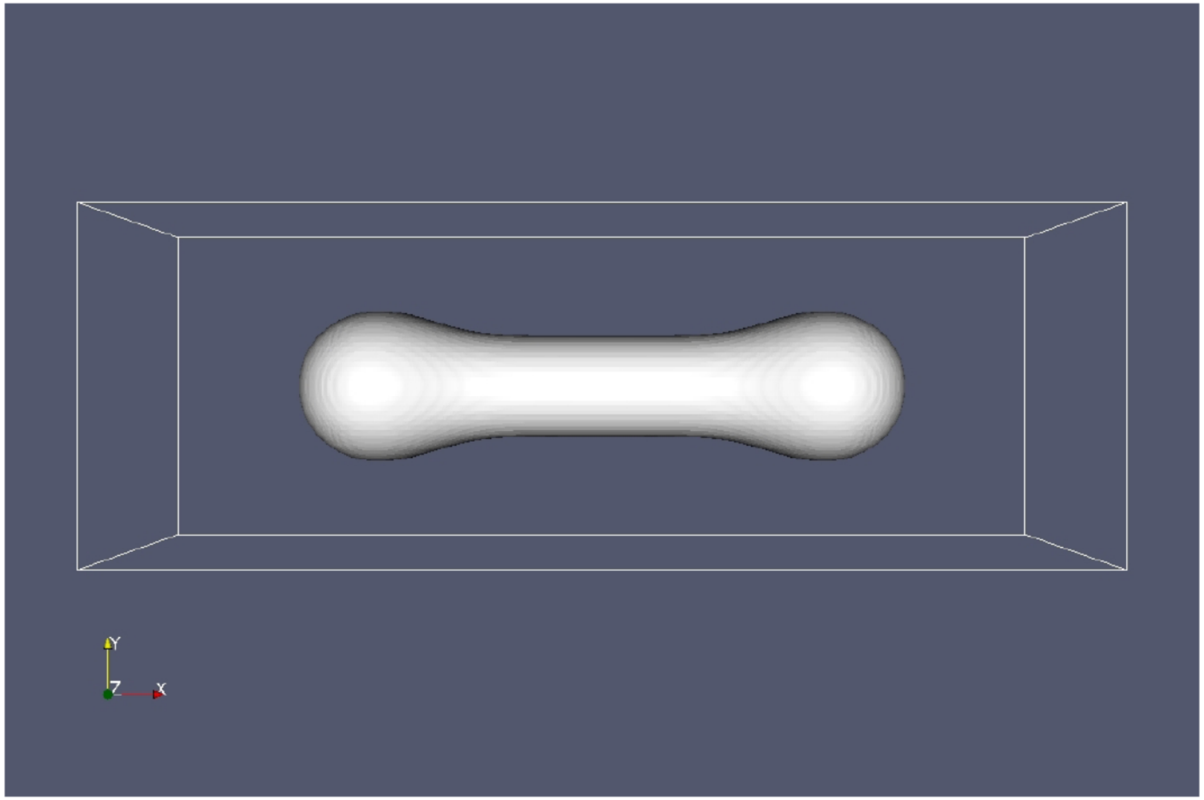
ΔP on painehäviö
η on nesteen dynaaminen viskositeetti
l on putken pituus
q_v on tilavuusvirta
d on putken halkaisija

Tehdaskoe

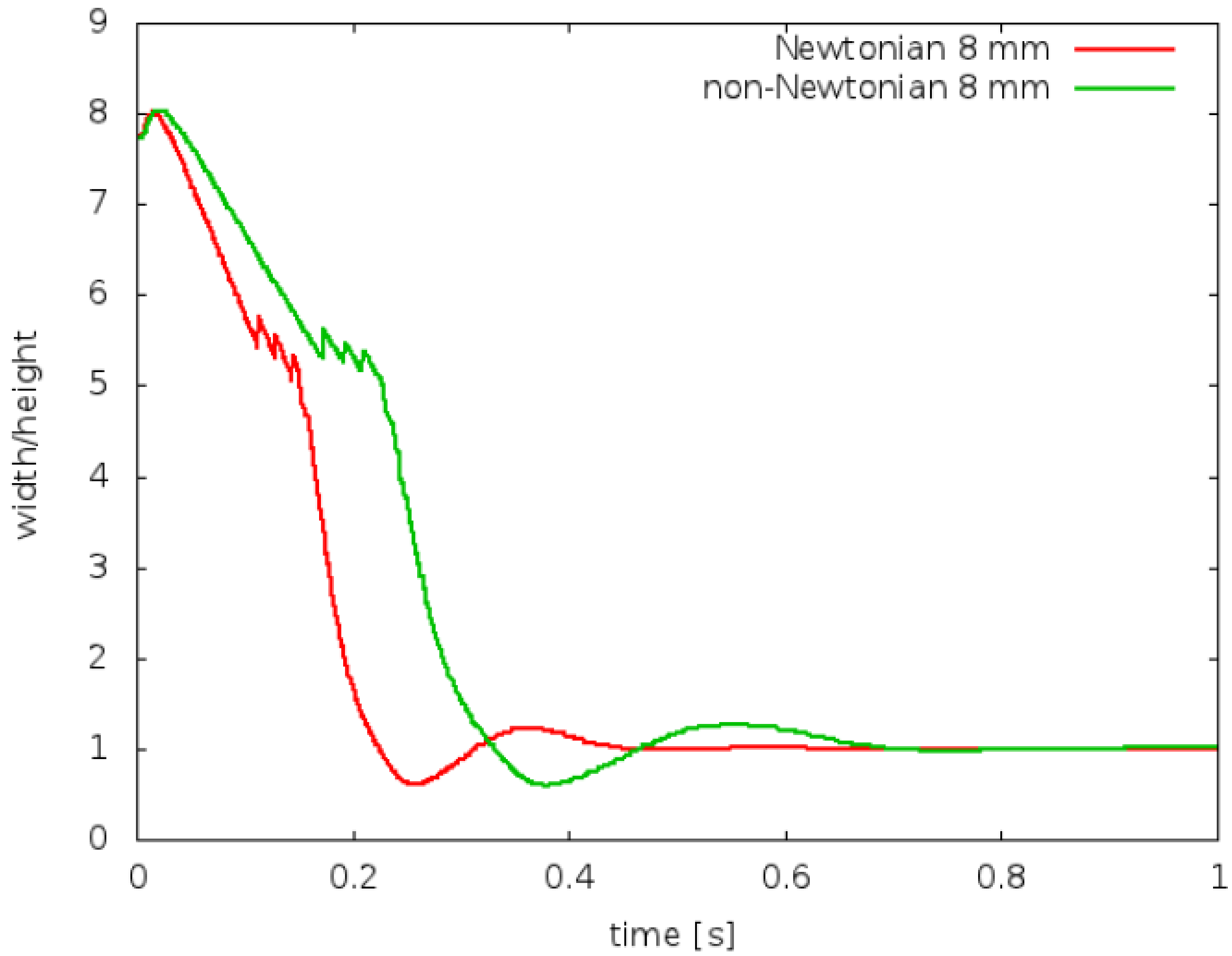


Black liquor (129°C)
Apparent viscosity [Pa·s]
0,14
0,12
0,1
0,08
0,06
0,04
0,02
0
0
100
200
300
400
500
600
700
800
Shear rate [1/s]

OpenFoam laskenta



Rihman ”vetäytyminen”



Seuraavaksi

- Analysoidaan havulipeän viskositeetti
- Näytteiden otto, 2 lipeää + Euka (Andritz)
- Mitataan viskositeetit laboratoriossa
- Simuloidaan 3-D pisarat (OpenFoam, Oulu)
- Julkaisu

LIITE II

Syöttövesipumppujen optimaalinen valinta
esitys 18.8.2013

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Syöttövesipumppujen optimaalinen valinta

Lappeenranta University of Technology
Prof. Esa Vakkilainen

Tarkasteltavat tapaukset

- Saatu dataa Suomesta neljästä kattilasta
  - Kaukopää SK6
  - Kaukopää SK5
  - Joutseno
  - Kymi
- Jokaisesta kattilasta kerätty vähintään vuoden ajalta tuntidata syöttövesivirtauksesta ja paineesta
- Kattiloiden toiminnan perusteella valittu tyypilliset toimintapisteet
- Saatujen SV-pumppujen datojen perusteella saatiin tyypillinen pumpun hyötysuhdekäyrä
- Investointikustannus perustuu saatuihin kustannuksiin (Wisa, Kymi)
- Pumppauskustannus perustuu kussakin toimintapisteessä pumppaustehon minmointiin valitsemalla sopiva märä pumppuja

Tarkasteltavissa tapauksissa pysyvyyskäyrät samanlaisia

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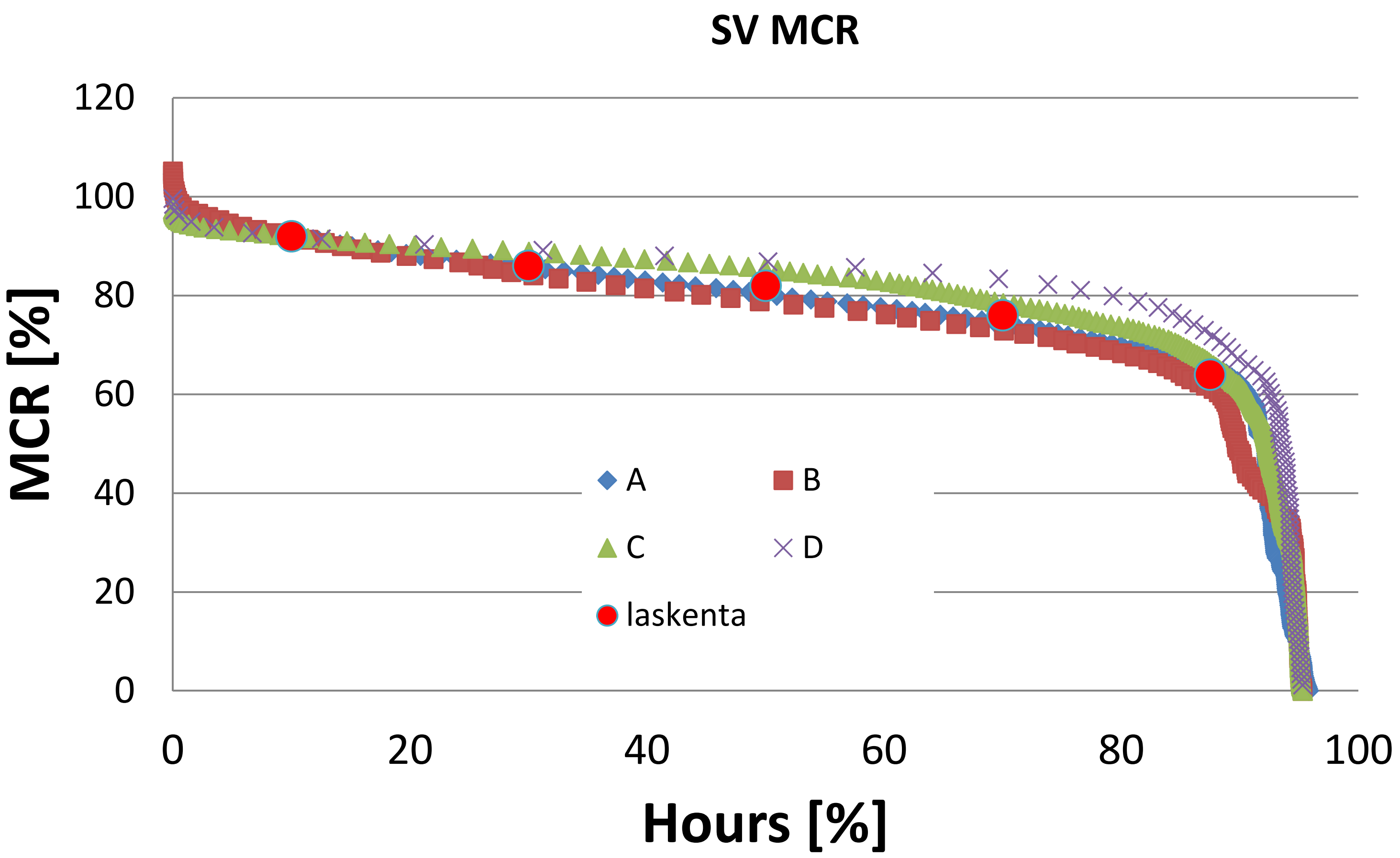
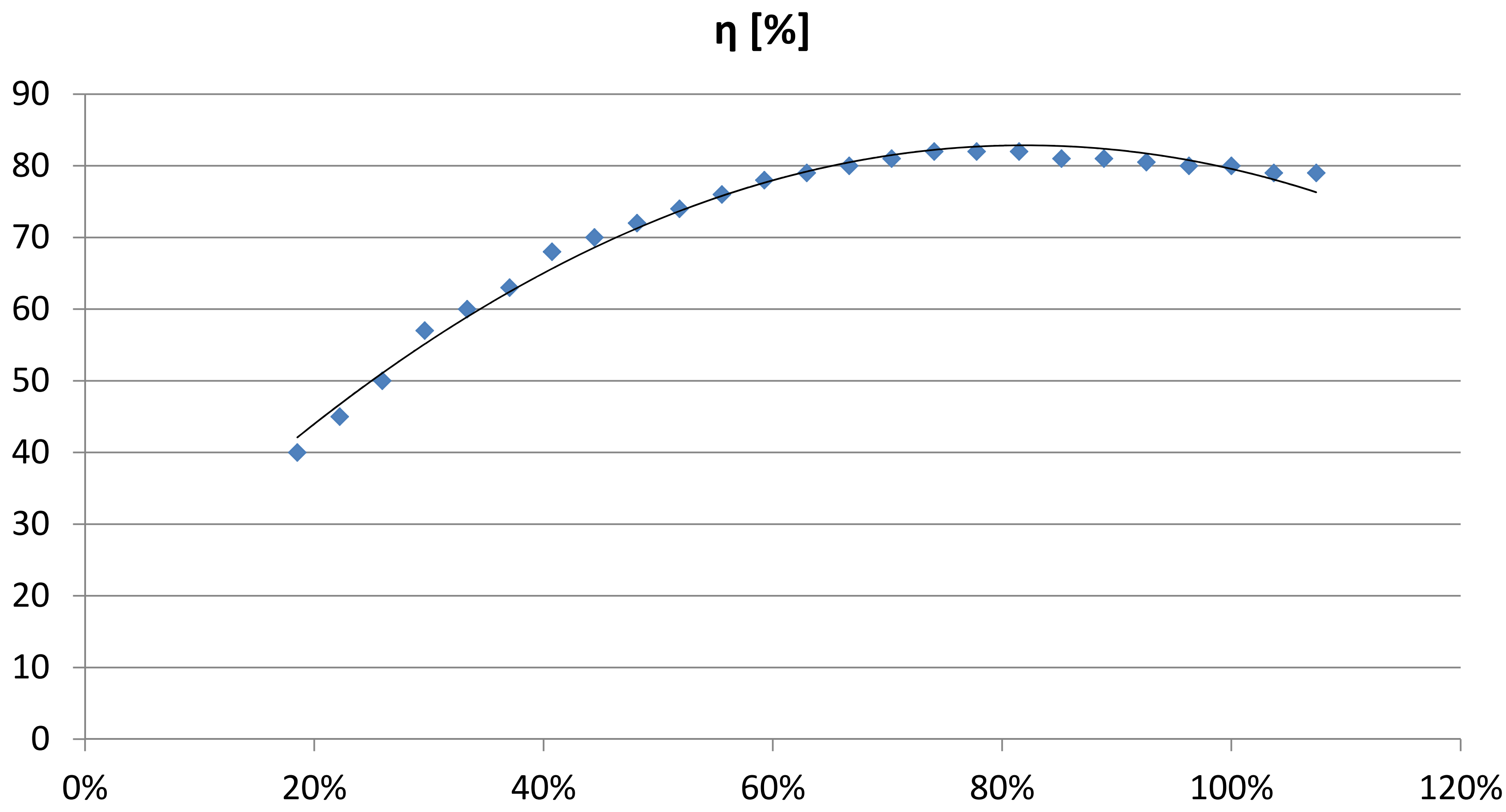
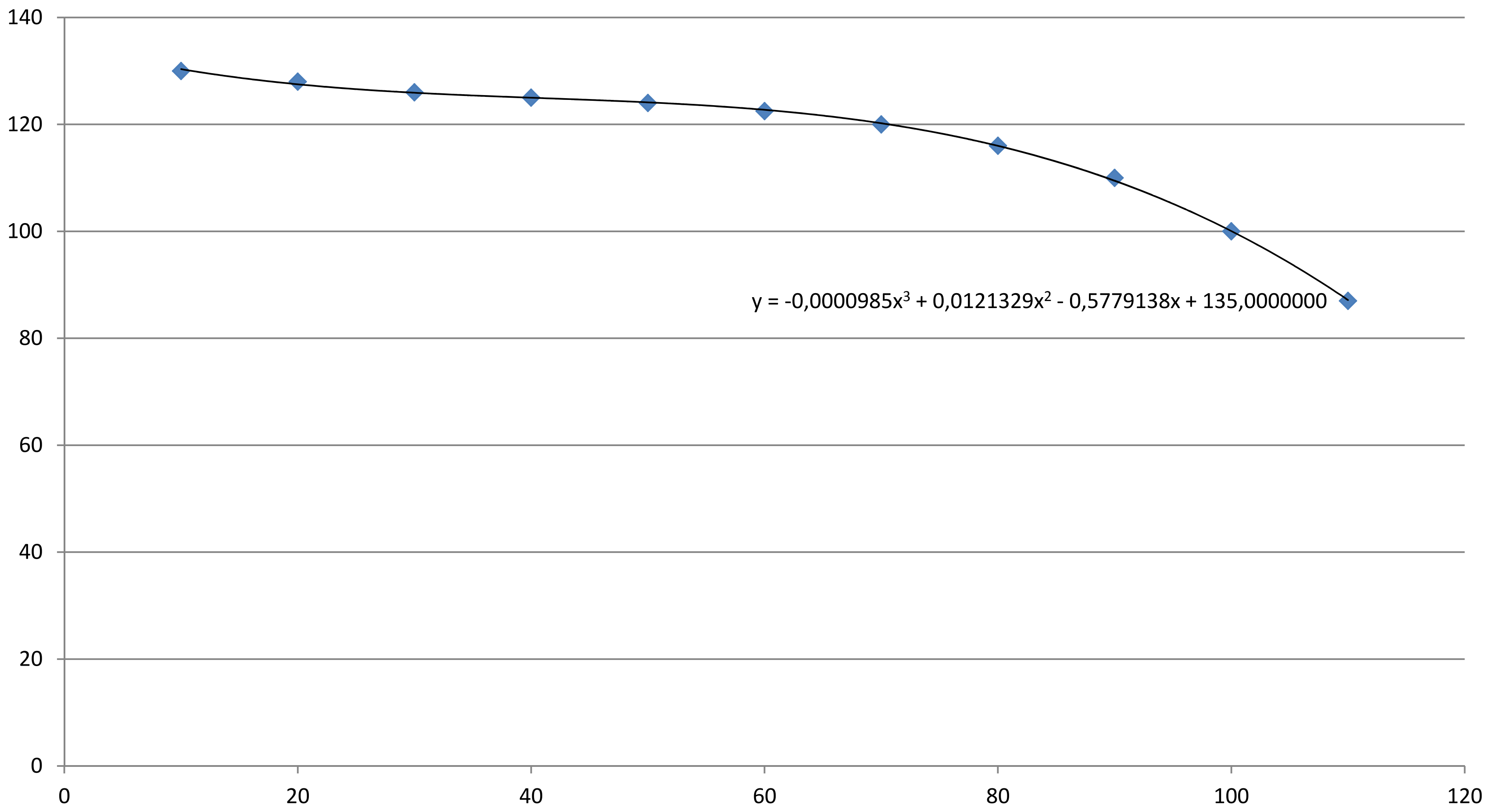


Tyypillinen pumpun hyötysuhdekäyrä



Tyypillinen pumpun nostokorkeus (kuristussäätö)



Valittiin tyypillinen tapaus

virtaus[MCR]	100	kg/s
paine sisään	0.5	MPa
paine ulos	10.3	MPa
SV-paineennousu	9.8	MPa
josta virtaushäviöt	1.2	MPa
tiheys	922	kg/m^3
sähkön hinta	40	€/MWh
ajoaika	8313	h/a
Moottorin η	92 %	

Laskettiin varmuuden vuoksi kaikki tapaukset (varalla turbopumppu)

	1					
käytössä olevat pumput	1*140					
Pumpun suunnittelu %MCR	168					
	2	1				
käytössä olevat pumput	2*70	1*70				
Pumpun suunnittelu %MCR	84	84				
	3	2	1			
käytössä olevat pumput	3*45	2*45	1*45			
Pumpun suunnittelu %MCR	54	54	54			
	4	3	2	1		
käytössä olevat pumput	4*33	3*33	2*33	1*33		
Pumpun suunnittelu %MCR	40	40	40	40		
	5	4	3	2	1	
käytössä olevat pumput	5*25	4*25	3*25	2*25	1*25	
Pumpun suunnittelu %MCR	30	30	30	30	30	
	6	5	4	3	2	1
käytössä olevat pumput	6*20	5*20	4*20	3*20	2*20	1*20
Pumpun suunnittelu %MCR	24	24	24	24	24	24

Laskelmaesimerkki

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	1	2	1	3	2	1	4	3	2	1
käytössä olevat pumput	**1*140**	**2*70**	1*70	**3*45**	2*45	1*45	**4*33**	3*33	2*33	1*33
Pumpun suunnittelu %MCR	168	84	84	54	54	54	40	40	40	40
0-1750 0-20 %	168			162			158.4			
Ajopiste mcr/pumppu	**92.0**	**46.0**	92.0	30.7	**46.0**	92.0	23.0	**30.7**	46.0	92.0
mcr/mcr_design	**0.55**	**0.55**	1.10	0.57	**0.85**	1.70	0.58	**0.77**	1.16	2.32
η [%]	**75 %**	**75 %**	75 %	76 %	**83 %**	31 %	77 %	**83 %**	71 %	31 %
Paine MCR	**98 %**	**90 %**	98 %	89 %	**90 %**	98 %	88 %	**89 %**	90 %	98 %
η,paine	**95 %**	**95 %**	92 %	95 %	**99 %**	78 %	97 %	**99 %**	96 %	66 %
kust. [k€]	**102**	**94**	106	91	**82**	302	88	**81**	99	357
1750-3500 20-40%										
Ajopiste mcr/pumppu	**86.0**	43.0	**86.0**	28.7	**43.0**	86.0	21.5	**28.7**	43.0	86.0
mcr/mcr_design	**0.51**	0.51	**1.02**	0.53	**0.80**	1.59	0.54	**0.72**	1.09	2.17
η [%]	**73 %**	73 %	**79 %**	74 %	**83 %**	31 %	75 %	**82 %**	76 %	31 %
Paine max [%]	**97 %**	90 %	**97 %**	89 %	**90 %**	97 %	88 %	**89 %**	90 %	97 %
η,paine [%]	**92 %**	90 %	**85 %**	92 %	**99 %**	82 %	94 %	**98 %**	92 %	68 %
kust. [k€]	**100**	95	**101**	90	**76**	265	87	**77**	90	319
3500-5250 40-60 %										
Ajopiste mcr/pumppu	**82.0**	41.0	**82.0**	27.3	**41.0**	82.0	20.5	**27.3**	41.0	82.0
mcr/mcr_design	**0.49**	0.49	**0.98**	0.51	**0.76**	1.52	0.52	**0.69**	1.04	2.07
η [%]	**72 %**	72 %	**80 %**	73 %	**82 %**	33 %	74 %	**81 %**	78 %	31 %
Paine max [%]	**96 %**	90 %	**96 %**	89 %	**90 %**	96 %	88 %	**89 %**	90 %	96 %
η,paine [%]	**90 %**	90 %	**90 %**	90 %	**98 %**	84 %	95 %	**98 %**	92 %	70 %
kust. [k€]	**99**	92	**88**	90	**74**	227	84	**74**	83	293
5250-7000 60-80 %										
Ajopiste mcr/pumppu	**76.0**	38.0	**76.0**	25.3	**38.0**	76.0	19.0	**25.3**	38.0	76.0
mcr/mcr_design	**0.45**	0.45	**0.90**	0.47	**0.70**	1.41	0.48	**0.64**	0.96	1.92
η [%]	**69 %**	69 %	**82 %**	70 %	**81 %**	48 %	71 %	**80 %**	81 %	31 %
Paine max [%]	**95 %**	90 %	**95 %**	89 %	**90 %**	95 %	88 %	**89 %**	90 %	95 %
η,paine [%]	**90 %**	90 %	**98 %**	90 %	**95 %**	88 %	90 %	**95 %**	94 %	72 %
kust. [k€]	**94**	88	**72**	86	**71**	138	85	**72**	72	261
7000-8313 80-95 %										
Ajopiste mcr/pumppu	**64.0**	32.0	**64.0**	21.3	**32.0**	64.0	16.0	21.3	**32.0**	64.0
mcr/mcr_design	**0.38**	0.38	**0.76**	0.40	**0.59**	1.19	0.40	0.54	**0.81**	1.62
η [%]	**63 %**	63 %	**83 %**	65 %	**78 %**	69 %	65 %	75 %	**83 %**	31 %
Paine max [%]	**93 %**	89 %	**93 %**	88 %	**89 %**	93 %	88 %	88 %	**89 %**	93 %
η,paine [%]	**87 %**	87 %	**96 %**	88 %	**93 %**	90 %	88 %	93 %	**98 %**	82 %
kust. [k€]	**87**	84	**61**	80	**64**	77	79	66	**57**	189
KUST. YHT [k€]/a	481	410		367			360			

	5	4	3	2	1	6	5	4	3	2	1
käytössä olevat pumput	**5*25**	4*25	3*25	2*25	1*25	6*20	5*20	4*20	3*20	2*20	1*20
Pumpun suunnittelu %MCR	30	30	30	30	30	24	24	24	24	24	24
0-1750 0-20 %	150					144					
Ajopiste mcr/pumppu	18.4	**23.0**	30.7	46.0	92.0	15.3	**18.4**	23.0	30.7	46.0	92.0
mcr/mcr_design	0.61	**0.77**	1.02	1.53	3.07	0.64	**0.77**	0.96	1.28	1.92	3.83
η [%]	79 %	**83 %**	79 %	31 %	31 %	80 %	**83 %**	81 %	62 %	31 %	31 %
Paine MCR	88 %	**88 %**	89 %	90 %	98 %	88 %	**88 %**	88 %	89 %	90 %	98 %
η,paine	97 %	**99 %**	90 %	67 %	45 %	98 %	**99 %**	90 %	85 %	48 %	32 %
kust. [k€]	86	**80**	93	320	523	84	**80**	90	126	452	736
1750-3500 20-40%											
Ajopiste mcr/pumppu	17.2	**21.5**	28.7	43.0	86.0	14.3	**17.2**	21.5	28.7	43.0	86.0
mcr/mcr_design	0.57	**0.72**	0.96	1.43	2.87	0.60	**0.72**	0.90	1.19	1.79	3.58
η [%]	77 %	**82 %**	81 %	45 %	31 %	78 %	**82 %**	82 %	69 %	31 %	31 %
Paine max [%]	88 %	**88 %**	89 %	90 %	97 %	88 %	**88 %**	88 %	89 %	90 %	97 %
η,paine [%]	95 %	**98 %**	90 %	76 %	48 %	92 %	**97 %**	95 %	92 %	77 %	34 %
kust. [k€]	84	**77**	85	184	453	85	**77**	79	98	262	639
3500-5250 40-60 %											
Ajopiste mcr/pumppu	16.4	**20.5**	27.3	41.0	82.0	13.7	16.4	**20.5**	27.3	41.0	82.0
mcr/mcr_design	0.55	**0.68**	0.91	1.37	2.73	0.57	0.68	**0.85**	1.14	1.71	3.42
η [%]	75 %	**81 %**	82 %	53 %	31 %	77 %	81 %	**83 %**	73 %	31 %	31 %
Paine max [%]	88 %	**88 %**	89 %	90 %	96 %	88 %	88 %	**88 %**	89 %	90 %	96 %
η,paine [%]	95 %	**98 %**	97 %	82 %	51 %	97 %	98 %	**98 %**	96 %	89 %	38 %
kust. [k€]	82	**74**	74	138	403	79	74	**72**	84	216	540
5250-7000 60-80 %											
Ajopiste mcr/pumppu	15.2	**19.0**	25.3	38.0	76.0	12.7	15.2	**19.0**	25.3	38.0	76.0
mcr/mcr_design	0.51	**0.63**	0.84	1.27	2.53	0.53	0.63	**0.79**	1.06	1.58	3.17
η [%]	73 %	**79 %**	83 %	63 %	31 %	74 %	79 %	**83 %**	77 %	31 %	31 %
Paine max [%]	88 %	**88 %**	89 %	90 %	95 %	88 %	88 %	**88 %**	89 %	90 %	95 %
η,paine [%]	90 %	**94 %**	98 %	34 %	56 %	91 %	93 %	**98 %**	88 %	74 %	40 %
kust. [k€]	82	**73**	67	258	336	80	73	**67**	80	240	470
7000-8313 80-95 %											
Ajopiste mcr/pumppu	12.8	16.0	**21.3**	32.0	64.0	10.7	12.8	16.0	**21.3**	32.0	64.0
mcr/mcr_design	0.43	0.53	**0.71**	1.07	2.13	0.44	0.53	0.67	**0.89**	1.33	2.67
η [%]	67 %	75 %	**82 %**	77 %	31 %	69 %	75 %	80 %	**82 %**	56 %	31 %
Paine max [%]	88 %	88 %	**88 %**	89 %	93 %	88 %	88 %	88 %	**88 %**	89 %	93 %
η,paine [%]	88 %	90 %	**95 %**	93 %	78 %	89 %	90 %	93 %	**97 %**	85 %	54 %
kust. [k€]	77	68	**59**	65	199	75	68	61	**57**	97	287
KUST. YHT [k€]/a	357					354					

Tulokset kuristussäätö (edullisimman määrän ajon mukaan)

INVESTOINTI					
Pumppu [kpl]	1	2	3	4	5
Mit. virtaus/pumppu [kg/s]	168	84	54	39.6	30
Nostokorkeus [MPa]	11.27	11.27	11.27	11.27	11.27
k€	202	249	274	294	303
Sähkömoottorit [kW]	3100	1500	1000	700	600
k€	272	245	246	240	258
Turbopumppu	2567	1283	825	605	458
k€	506	312	229	184	152
Asennus ym					
k€	142	197	244	286	325
YHT	1290	1087	1047	1044	1068
INV. KUST. YHT [k€]/a	184	155	150	149	153
PUMPPAUS [k€]/a	684	609	593	593	588
KOKONAISKUST. [k€]/a	868	764	742	742	741

Tulokset nestekytkin (edullisimman määrän ajon mukaan)

INVESTOINTI					
Pumppu [kpl]	1	2	3	4	5
Mitoitus virtaus/pumppu [kg/s]	168	84	54	39.6	30
Nostokorkeus [MPa]	11.27	11.27	11.27	11.27	11.27
k€	202	249	274	294	303
Sähkömoottorit [kW]	3100	1500	1000	700	600
k€	272	245	246	240	258
Turbopumppu	2567	1283	825	605	458
k€	506	312	229	184	152
Nestekytkin [kW]	3400	1700	1100	800	700
k€	290	357	395	421	480
Asennus ym					
k€	112	148	179	207	233
YHT	1382	1311	1323	1347	1425
INV. KUST. YHT [k€]/a	197	187	189	192	204
PUMPPAUS [k€]/a	543	420	404	404	381
KOKONAISKUST. [k€]/a	740	607	593	597	585

Tulokset invertteri (edullisimman määrän ajon mukaan)

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INVESTOINTI					
Pumppu [kpl]	1	2	3	4	5
Mitoitus virtaus/pumppu [kg/s]	168	84	54	39.6	30
Nostokorkeus [MPa]	11.27	11.27	11.27	11.27	11.27
k€	202	249	274	294	303
Sähkömoottorit [kW]	3100	1500	1000	730	550
k€	272	245	246	247	243
Turbopumppu	2567	1283	825	605	458
k€	506	312	229	184	152
Invertteri [kW]	3400	3300	3300	3200	3100
k€	304	367	415	442	463
Asennus ym					
k€	142	197	244	286	325
YHT	1427	1370	1408	1454	1486
INV. KUST. YHT [k€]/a	204	196	201	208	212
PUMPPAUS [k€]/a	481	410	367	360	357
KOKONAISKUST. [k€]/a	685	606	568	568	569

Vertailu 2 tai 3 pumpulla

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INVESTOINTI	Kuristussäätö		Nestekytkin		Inventteri	
Pumppu [kpl]	2	3	2	3	2	3
Mitoitus virtaus/pumppu [kg/s]	84	54	84	54	84	54
Nostokorkeus [MPa]	11.27	11.27	11.27	11.27	11.27	11.27
k€	249	274	249	274	249	274
Sähkömoottorit [kW]	1500	1000	1500	1000	1500	1000
k€	245	246	245	246	245	246
Turbopumppu	1283	825	1283	825	1283	825
k€			312	229	312	229
Invertteri [kW]			1700	1100	3300	3300
k€			357	395	367	415
Asennus ym						
k€	197	244	148	179	197	244
YHT	1087	1047	1311	1323	1370	1408
INV. KUST. YHT [k€]/a	155	150	187	189	196	201
PUMPPAUS [k€]/a	609	593	420	404	410	367
KOKONAISKUST. [k€]/a	764	742	607	593	606	568

Tulokset

- Näyttää siltä että **kolme pumppua ja turbopumppu** on kaikissa tapauksissa edullisin tai ainakaan eroa neljä pumppua ja turbopumppu on pieni.
- Kaksi pumppua ja turbopumppu on aina kallis vaihtoehto
- Yksi pumppu ja turbopumppu on aina kallein vaihtoehto
- Syöttövesipumppujen teho on ylimitoitettu osittain standardin vaatimuksesta


I'M A LUMBERJACK AND I'M OUKEI!
seppo.net

LIITE III

Bahnam Zakri, Optimal Design of Boiler Feed Water Pumping System: Simulation and Techno-economic Analysis in Variable Load, Master's thesis

LAPPEENRANTA UNIVERSITY OF TECHNOLOGY

Faculty of Technology

Department of Energy Technology

Master's Thesis

Bahnam Zakri

OPTIMAL DESIGN OF BOILER FEED WATER PUMPING SYSTEM:

SIMULATION AND TECHNO-ECONOMIC ANALYSIS IN VARIABLE LOAD

Examiner: Professor, D.Sc. Esa Vakkilainen
Docent, D.Sc. Juha Kaikko

Supervisor: Professor, D.Sc. Esa Vakkilainen

Lappeenranta, 2013

ABSTRACT

Lappeenranta University of Technology

Faculty of Technology

Department of Energy Technology

Bahnam Zakri

Optimal Design of Boiler Feed Water Pumping System: Simulation and Techno-economic Analysis in Variable Load

Master's thesis

90 pages, 30 figures, 13 tables and 2 appendices

Examiners: Professor, D.Sc. Esa Vakkilainen
Docent, D.Sc. Juha Kaikko

Keywords: boiler feed water pump, energy efficiency, simulation, optimization, variable speed pumping, steam power plant

Boiler feed water pumps are among the most energy-intensive auxiliary units in steam boiler plants. While maintaining high availability and reliability has traditionally dominated the optimized design of these pumps, they are good candidates for enhancing the energy efficiency of the plant, from system design viewpoint. This study seeks to establish an algorithm to optimize the life cycle costs of the boiler feed water pumping system in a variable load scheme. Using MATLAB programming tool, a holistic mathematical model is built for analysis, computation, and optimization of various pumping systems, including single and multiple pumps in constant or variable speed. IPSEpro process simulation tool is then employed to study the performance of each pumping solution in a medium-sized power plant, using real load data of the plant in off-design operation. Using this approach, net electrical efficiency and carbon emissions of the plant in different loads is compared under the application of each pumping system.

ACKNOWLEDGEMENTS

The achievement of this study is the result of knowledge provided by teachers and lecturers in my entire study life, to whom I always thank. I would like to thank prof. Vakkilainen, the supervisor of this study, for his helpful comments, rich ideas, and continuous support. He is always quick in response and available to meet, which is valuable for a short-term, laborious research task.

There are many persons to whom I should thank for their contribution to this study. Many thanks to Juha Viholainen, he gave me good ideas and comments regarding pumping systems and their simulation in MATLAB. I also thank D.Sc. Juha Kaikko for teaching IPSEpro tool and other helps during studies. My sincere thanks are dedicated to D.Sc. Tommi Ekholm from VTT for dedicating time and helping me in performing optimization tasks. I should also state my appreciation to prof. Risto Lahdelma from Aalto University for accepting me in courses relevant to this topic and providing links to useful experts in this subject.

I should thank EFEU program at CLEEN Ltd. for supporting this study, as well as Lappeenranta University of Technology for providing funds. I also appreciate Edward Paro from Sulzer Pumps Finland and Jukka Tolvanen from ABB Finland for their efforts to provide information about pumping systems. I cannot finish my acknowledgements without remembering my colleagues and friends who provided a nice environment for doing the task at LUT.

I state my warm thanks to Khakpoor family, Lina Teir and her family for their help and supports during my stay in Finland. The biggest thanks belong to my family, particularly to my mother who has been always inspiring me for being hardworking and determined.

TABLE OF CONTENTS

ABBREVIATION

aux	auxiliary
ASD	adjustable speed drive
bd	blowdown
BEP	best efficiency point
BFP	boiler feed water pump
BFPS	boiler feed water pumping system
BHP	brake horsepower
CHP	combined heat and power
CPI	consumer price index
CSP	constant speed pump/pumping
daf	dry-ash-free basis
desup	desuperheating
ECO	economizer
EN	European Standard
FWH	Feed water heater
gpm	(US) gallon per minute
HP	high pressure
hp	horsepower
LHV	lower heating value
LP	linear programming (linear optimization)
MILP	mixed integer linear programming
ms	multistage (pump)
NPSH	net positive suction head
NPSHa	net positive suction head available
NPSHr	net positive suction head required
NPV	net present value
SH	superheater
TH	total head
USCS	United States Customary System
VFD	variable frequency drive
VSD	variable speed drive
VSED	variable speed electric drive
VSP	variable speed pump/pumping
WHP	water horsepower

SYMBOLS

g	gravity acceleration [m/s^2]
H_f	friction head loss [Pa]
$H_{f,p}$	piping friction head loss [Pa]
$H_{f,vf}$	valve/fitting friction head loss [Pa]
$h_{f,l}$	relative head loss [-]
K	valve/fitting resistant coefficient [-]
N	rotational speed [rpm]
n_q	pump specific speed [-]
n_s	suction specific speed [-]
P	power [W]
P	pressure[Pa]
Q	flow rate [m^3/s]
r	discount/interest rate [%]
s.g	specific gravity[-]
V	velocity [m/s]
Z	elevation [m]
η	efficiency [%]

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1. INTRODUCTION

Pumping systems account for 20% of the world's electricity consumption and up to half of the energy use in some industrial plants [1]. In steam power plants, the electrical energy consumed by the boiler feed water pump (BFP) is approximately 2 to 5% of the total generated electricity, up to several megawatts based on the plant size [2]. Optimized selection and control of BFPs can reduce their energy consumption resulting in higher net energy efficiency and less carbon emissions for the steam generating plants. However, there are several challenges in mitigation of energy losses of BFPs.

Boiler feed water pumping system (BFPS) is truly considered the heart of the steam generating plants. Since any failure or service outage results in heavy production losses, reliability and availability of BFPS must be carefully ensured. Pump engineers therefore tend to calculate and design the BFPS more conservatively to respond the probable future increases in the duty. Not only pump engineers, boiler standards also require relatively high safety margins that must be covered by the selected BFP, resulting in oversizing of the system. Moreover, the increasing use of intermittent renewable energy sources in power systems has imposed more load fluctuations to the central power stations. It further leads the BFPS to the part load operation, relatively far from the initial design point.

To address the mentioned challenges, the BFPS should be optimally designed for the entire life cycle. The optimization can be performed for the energy costs or more precisely for the total life cycle costs (LCC). Having the load pattern of the plant on a yearly basis can contribute the optimized selection of the new BFP when the existing system is retrofitted or replaced. In this study, the BFPS is optimally designed for a medium-size steam power plant, benefiting from the actual load data of an operating plant in Finland.

In the first scenario, a single constant-speed BFP is compared with variable speed and multiple pump alternatives to examine the LCC for each design. Next, a BFPS is examined to achieve the optimized control strategy for the given load regime in operation. The system comprises a booster pump and two identical parallel BFPs. MATLAB programming tool is used for building the mathematical model and optimization of the various pumping systems. To monitor the performance of the steam power plant using each BFPS, the whole plant is modeled and simulated with IPSEpro simulation tool. Since some of the power losses in BFPs are absorbed by the pumped water in terms of heat, it affects the net energy efficiency of the plant in each pumping system. Having whole the plant simulated, such influences can be more precisely monitored. The empirical part of this study is limited to constant pressure drum boilers while sliding pressure is addressed in the theoretical part.

1.1. Background

Ever-rising energy prices and more stringent environmental regulations are strong motives to enhance the energy efficiency of the steam generating plants and reduction of in-house energy consumption. If not the most, BFPs are among the most energy consuming auxiliary units in boiler plants. BFPs are used to maintain the desirable boiler pressure and consequently the outgoing live steam pressure. Any fluctuation in load results in change in boiler pressure. It deviates the pump's operating point from the initial design point. To respond the load changes, the discharge pressure of the BFPs are traditionally regulated by a throttling valve or bypassing a share of the outlet stream. Using such approaches introduces significant losses in the system resulting in energy waste and faster wearing of the pump itself.

To reduce the losses, the operating point of the pump should be moved towards operating point of the system, for instance, by reduction of pump's rotational speed. Though the use of variable speed pumping (VSP) is not a new approach, the recent technological advancements in variable (adjustable) speed electric drives (VSED, ASD or simply VSD) has offered more economical achievements in VSP. In VSP systems, the reduction in driver's speed results in decrease in pump speed offering power savings in lower loads. While VSD technology has proved to be an economic option for low static head pumping systems, yet its usage in pumping applications with relatively high static head should be studied for each case [3].

BFPs are characterized for high static heads. However, continuous operation in part load and relatively high number of working hours offers further examination of VSP in BFPs. This study seeks to address this question. Though there is good literature in VSP in pumping stations and closed systems, yet the application of variable speed in BFPS and optimization of the whole system based on variable load data is not a properly covered topic.

1.2. Objectives and Scope of the Study

In this thesis, the author strives to find a theoretically defendable and practically acceptable approach for optimal design of the BFPS. Here, optimization objective is minimizing the life cycle costs of the BFPS, but minimization of energy consumption is also covered. Optimization is examined in two different steps: “selection” and “operating control”. First, BFPS is optimally selected among a set of available alternatives based on the available load data. In this approach, selection of a specific pump from a set of proposed BFPs is one criterion. Then, the number of main pumps in parallel, the stage number of each pump, and the best option among different available booster pumps are also examined. Based on “system design” concept, the optimized solution is determined for constant and variable speed schemes.

In the second optimization practice, the selected system is examined to find the optimized controlling strategy when two identical BFPs are in parallel. It means finding the best plan for running one or two pumps with constant or variable speed in different loads. In this section, optimization based on the energy efficiency of the BFPs is also briefly covered. However, the study remains limited to two identical pumps in this part to avoid complexity and cumber.

The monitoring of the performance of the each selected BFPS in a real power plant is another objective in this study. The application of BFPs in a thermal cycle of a combined heat and power (CHP) plant is also important in study of the total output of the plant in different loads. To do so, a medium-sized constant-pressure drum boiler plant is modeled and simulated in IPSEpro simulation environment. Using IPSEpro simulation tool, the net energy efficiency of the plant and turbine heat rate are determined under each pumping system. Moreover, the CO_2 emissions of the CHP plant are also determined for a biomass-fueled boiler.

1.3. Outline of the Thesis

Following the introduction in the first chapter, the basics of centrifugal pumps and their associated topics are reviewed in the second chapter. The focus is merely given to the topics related to the hydraulic performance characteristics rather than pump’s configuration and fluid dynamics. The required literature to understand the methods for calculation of main features of the pump, like power and efficiency, is also introduced. At the end of this chapter, a brief review of the pump configuration is added to give the general understanding of the typical BFP's configuration.

In chapter 3, the selection criteria and sizing of the centrifugal pumps is discussed. The attention is mostly paid to the variables related to system design and more specifically for boiler applications. System head curve and pumping control strategies for maintaining the system requirements in different load is also reviewed. The application of multiple pumps and their influence on the pumping output, as well as different pumping drives and transmission devices are studied in this chapter. LCC analysis and energy efficiency considerations are other parts of this chapter to facilitate the reader with required knowledge in study of life cycle or energy optimization in pumping systems.

Chapter 4 directly deals with BFPs, from system characteristics to relevant standards and driving options. The main parameters in determination of the operating point of the BFP are examined. The difference in pressure regulating between two major boiler types is adequately discussed, i.e. sliding pressure in once-through boilers and constant pressure in drum boilers. Since some of the topics are generally reviewed in the previous chapters, the goal is to highlight the main differences of the BFPS from the others, not to discuss them in details. At the end, a brief explanation of the system based approach is given and influential factors in study of a BFPS are reviewed.

The empirical part of the thesis is introduced in chapter 5. After a short discussion of the optimization problem, different scenarios in optimal design of BFPS are examined. Selection of one single unit from a set of alternatives is the first scenario. Next, the number of multiple BFPs is optimized to minimize the LCC. Then, optimization is performed to determine the best strategy in operational control of two identical parallel pumps in different loads. Finally, different optimized solutions are simulated in a real power plant model to compare the outcome of the plant in each scenario. Last but not the least, results are discussed and a conclusion is drawn in the last chapter, chapter 6.

2. CENTRIFUGAL PUMPS

Pumps are divided to two major categories based on the principle by which energy is transferred to the liquid, positive displacement and kinetic. In positive displacement pumps, energy is periodically transferred to movable volumes of the liquid by direct application of a forcing device like reciprocating piston. This type of the energy addition directly increases the pressure of the liquid at the discharge line to the designated level. Reciprocating and rotary pumps are the most common sub-categories in the category of positive displacement pumps [4].

In kinetic pumps, energy from the pump driver is used to continuously increase the velocity of the liquid. Then, reduction in the velocity leads to a pressure increase as the goal of pumping process, based on the Bernoulli's equation. For the major part, kinetic pumps consist of centrifugal pumps besides several special types, e.g. regenerative turbines and special effect pumps. This second division is made based on the means by which the energy addition is applied. As the centrifugal pumps are the most common pump types, particularly for BFP applications, they are examined more detailed in this chapter. The other pump types are not further considered to avoid unnecessary details.

2.1. General Characteristics

In general, a centrifugal pump consists of three main components: shaft, impeller connected and rotated by the shaft, and casing (Fig.2.1).

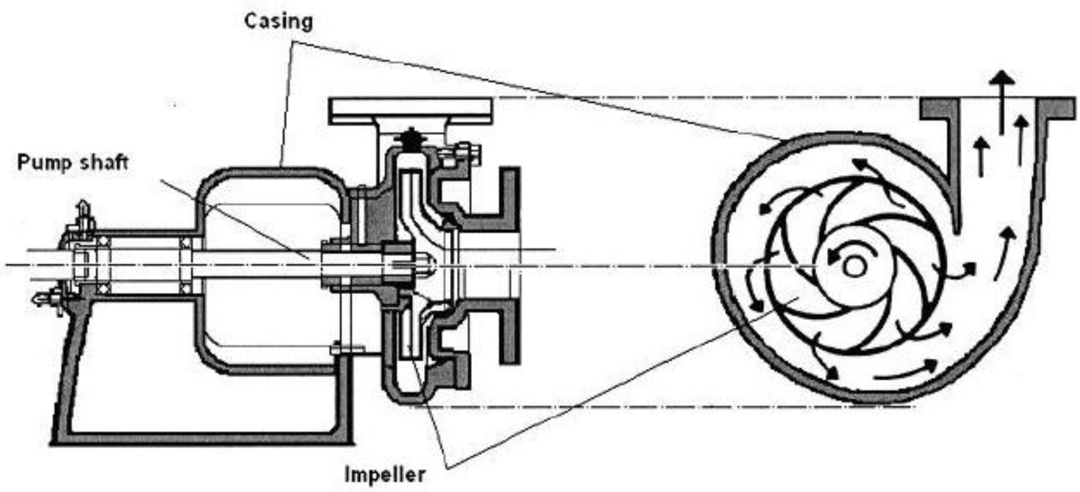


Figure 2.1 Main components of a centrifugal pump [5]

The impeller is attached to the shaft and liquid is forcing to move towards the discharge side by the rotation of the impeller. The consequent void (reduced pressure area) at the impeller inlet is then filled by the higher pressure at the pump casing inlet (suction pressure). Once the liquid enters the pump, it is directed to the impeller vanes resulting in velocity increase. Leaving the impeller outlet vane tip, the liquid reaches the maximum velocity. The kinetic energy of this velocity rise is transformed into pressure increase through a diffusion process occurring in the pump casing (Bernoulli Equation) [6]. The expansion in cross-sectional area of the casing ensures this pressure increase.

2.2. Pump Hydraulics and Curves

In this section, the basics of pump hydraulics are briefly discussed to provide adequate literature for the following parts. The focus is given to practical subjects related to the main requirements for sizing and selection of pumping systems, particularly centrifugal pumps. Performance curves are then introduced and the usage of each is briefly discussed.

2.2.1. Head and Capacity

Pumping systems are used to increase the pressure or head of a liquid flow. Head or total head (TH), usually expressed in meters or feet, is a differential pressure that is produced by pumping operation. The amount of produced head by a specific pump is constant at a certain flow rate, if the pump speed and impeller size remains intact. In other words, the produced head is a function of flow arte, or capacity, and varies with the flow rate handled by the pump. Pump capacity is usually expressed in liter per second (l/s), cubic meter per hour (m^3/hr), or for the larger pumps cubic meter per second (m^3/s). The equivalent in US customary system (USCS), depending on the pump size, is gallon per minutes (gpm), cubic feet per minute (cfm), or cubic meter per second (cfs).

2.2.1.1. Head

In general, based on the Bernoulli's Equation, the head developed by a pumping plant can be expressed as:

$$H_{pump} = \frac{P_{out} - P_{in}}{\rho g} + Z_{out} - Z_{in} + \frac{V_{out}^2 - V_{in}^2}{2g} \qquad (2.1)$$

H_{pump} = head developed by pump [m]

P_{in} , P_{out} = inlet and outlet pressure of the pump [Pa]

ρ = liquid density [kg/m^3]

g = acceleration of gravity [m/s^2]

Z_{in} , Z_{out}= elevation at suction and discharge [m]

V_{in} , V_{out} = liquid velocity at suction and discharge of the pump [m/s]

In principle, a centrifugal pump increases the flow velocity and then converts some of this velocity into pressure through a diffusion process. The amount of developed head by the pump is related to the velocity:

$$H = \frac{{V_{im}}^2}{2g} \qquad (2.2)$$

H = head [m]

V_{im} = velocity at the tip of the impeller [m/s]

g = gravity acceleration [m/s^2]

The impeller diameter and the velocity at the tip of the impeller are related:

$$V_{im} = r\omega \qquad (2.3)$$

V_{im} = velocity at the tip of the impeller [m/s]

ω = rotational speed [rad.s^{-1}]

r = impeller radius [m]

Comparing the equation 2.2 and 2.3, the linear velocity can be eliminated for head:

$$H = \frac{(r\omega)^2}{2g} \qquad (2.4)$$

Based on the equation 2.4, the head developed by the pump is merely a parabolic function of rpm and impeller diameter. The other important fact is that the head (expressed in units of meters or feet) is not a function of specific gravity of the pumped liquid. It means a pump moving a liquid up to a static distance of 100 m always has a head of 100 m, regardless of the gravity of pumped liquid. If liquids with different densities would be pumped by one pump with certain amount of head, different end pressures and power consumption for each liquid are observed. The relationship between pressure and head can be expressed as follows:

$$P = \rho g H \qquad (2.5)$$

P = pressure gauge at pump discharge [Pa]
H = head [m]
ρ = fluid density [kg/m^3]
g = acceleration gravity [m/s^2]

Therefore, it is important to convert the pressure units to the corresponding units of head (meters or feet etc.) before dealing with pump curves. It should be reminded that although head is theoretically specific energy and pressure is force over an area, they can be interchangeable terms in the gravitational field of the earth. As pumping systems studied in this work are near the earth surface, these terms are used interchangeably hereafter.

In order to calculate the suitable size of a centrifugal pump for a specific application, all the components existing in the system that introduce demand for head should be considered. In general, there are four different components for the system head referring to the liquid pressure, including:

1. Elevation head (H_z)
2. Pressure head (H_p)
3. Friction head (H_f)
4. Velocity head (H_v)

It is important to remember that the definition and name of these terms is slightly different in literature. In this study, the usage of these terms is based on the definition provided in this section. The sum of these four components results in total head (H_{tot}), which is the head that pump should overcome.

$$H_{tot} = H_z + H_p + H_f + H_v \qquad [m] \qquad (2.6)$$

A brief explanation of each component is provided in the following sections.

2.2.1.2. Elevation head

Elevation head is the total elevation difference that the liquid must be pumped. In general, elevation head is normally calculated by difference from the surface level of the liquid at the supply vessel to the upper level in the vessel where the liquid is pumping to. For instance, for a boiler pumping system, this head is difference in elevation of water surface from deaerator to the drum water level. In calculation of elevation head, the location of pump is not influential in the required head. If the supply vessel would be in a higher level than pump, there is a *suction head* in the system while having liquid level below the pump centerline introduces *suction lift* to the system [6]. If the pressure value is given by reading of gauge pressures at two different points in the suction and discharge piping of the pump, the elevation head is then the elevation difference between the two gauges, not the liquid levels at supply and delivery vessels.

2.2.1.3. Pressure head

Pressure head is the head needed for maintaining a pressure at upstream of the pump or overcome a vacuum at downstream. Pressure head is usually measured as the difference between the liquid surface pressure in the supply and delivery vessels. There is no pressure head in systems in which the pressure of supply and delivery vessels is the same, e.g. two atmospheric tanks. In this case, the term of pressure head is not considered in calculation of total head (eq.2.6). There is also no pressure head in closed loop single-phase systems without taking work out of the liquid, e.g. water circulating systems. In case suction vessel would be under vacuum conditions, the equivalent amount of vacuum in gauge pressure must be added to the delivery vessel gauge pressure, both converted in meters, in calculation of total head. It should be noted that pressure head is usually the main term in determining BFP total head. The combination of elevation head and pressure head is usually known as *static head*.

2.2.1.4. Friction head

Friction head is equal to the friction losses in the piping, fittings and valves that should be overcome by the operation of pump in the system. In a piping system, friction losses are related to the square of the velocity of the pumped liquid in a fully turbulent flow. There is a direct correlation between piping size and friction head loss. Smaller size of the pipe, valves, and fittings for a specific flow rate, introduces larger friction head loss for the system though the piping cost is lower. In other words, by reducing the costs of piping system by means of designing smaller sizes, the cost of pumping system including pump, driver and accessories increases due to the rise in friction head loss. It also increases lifetime energy costs of the pumping systems for the larger size. By reducing some complicated formulae to tables and charts friction losses can be calculated for pipes, valves and fittings. Using these tables for a specific piping material and size, the friction head loss can be calculated through a piping system including pipe, valves and fittings:

$$H_f = H_{f,p} + H_{f,vf} \tag{2.7}$$

$$H_{f,p} = h_{f,l} \times \frac{L}{l} \tag{2.8}$$

$$H_{f,vf} = K\frac{V^2}{2g} \tag{2.9}$$

H_f = friction head loss [m]

$H_{f,p}$ = piping friction head loss [m]

$H_{f,vf}$ = valve/fitting friction head loss [m]

$h_{f,l}$ = head loss coefficient per l units of piping linear length (pipe table)

L = piping length [m]

K= valve/fitting resistant coefficient given by the relevant chart or table [-]

$\frac{V^2}{2g}$ = velocity head given in the pipe table [m]

The total friction head loss is determined by summing up of all the friction losses of piping components in discharge and suction of the pump. It should be reminded that liquid flow rate and pipe size are two determining factors in calculation of friction losses. Friction losses are important to consider as they can cause head losses in suction line resulting in cavitation (further discussion in section 2.2.4). In boiler pumping systems, friction losses are not very large compared to pressure and static head but subject to study. The amount of friction losses increases in higher flow rates (higher velocities) resulting in a slight increase in boiler system head.

2.2.1.5. Velocity head

Velocity head is the change in the energy of a liquid in motion as a result of its velocity. The amount of velocity head is given for different pipe materials for different flow rates and diameter sizes. There are always two set points in suction and discharge side for calculation of different components of total head. Velocity head can be included in the total head calculations if the pressure head requirements are introduced as gauge readings at some points in the suction and discharge piping system (having velocity at set points). It appears in the same format expressed in equation 2.2. If set points for calculation of head for a system are chosen from the liquid level in supply and delivery vessels, the velocity head is zero as there is no velocity in these vessels.

In other cases in which velocity head component should be included in the head calculations, it can be expressed as the change in velocity head from suction to discharge of the pump. These values can be directly read from pipe tables. As the suction pipe size is typically bigger than discharge, velocity head is smaller, but not significantly. In boiler application, velocity head can be neglected if the head calculations are performed between deaerator and drum vessels (zero velocity).

2.2.1.6. Capacity

The required capacity at which the pump operates is normally dictated by the system requirements in which the pump is working. A specific process system is designed for a particular flow rate to meet its target. For instance, the steam production rate in the boiler, which is dictated by turbo-generator, specifies the amount of feed water needed for continuous operation. Despite of design operating point of the pump, it is usually possible to arrive at different flow rates dictated by the process nature. As mentioned before, since the pump head is dependent on its capacity, increase in the capacity results in lower heads and vice versa (Fig.2.2). This relationship is further discussed in next sections.

2.2.2. Performance Curves

The characteristic head–capacity curve shown in figure 2.2 is considered for a centrifugal pump operating at constant speed suitable for the pumping application. However, centrifugal pumps are usually capable to operate in a wider range of head and flow, even at a constant speed. This is possible by trimming the impeller diameter from its maximum size to a predefined minimum size, without modifying the pump casing and configuration. The minimum trim size is defined and suggested by the manufacturer. A centrifugal pump produces a group of head-capacity performance curves for a specific pump speed based on impeller diameter size (figure 2.2).

The upper curve in this envelope indicates the maximum diameter size that fit into the pump casing and the lower boundary represents the minimum impeller size offered by the pump manufacturer for economical operation of this particular pump. The right end of each curve shows the maximum flow that can be handled with this pump while the left end of the curves indicate the lowest possible flow for having pump in operation, called *shutoff* point of the pump. Based on the suction and discharge size, maximum impeller diameter and pump speed, manufacturers usually publish a set of envelopes for a particular pump configuration.

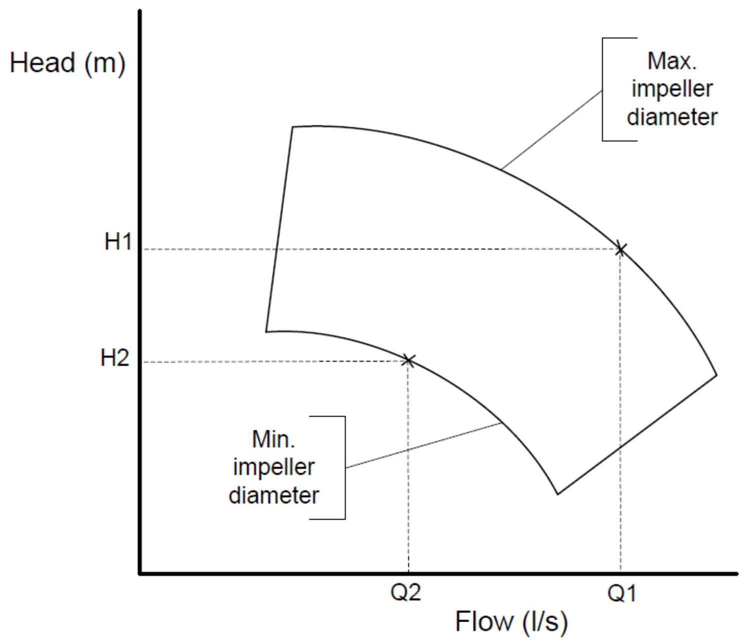


Figure 2.2 Head-capacity envelope in constant speed for a typical centrifugal pump

For instance, the pump illustrated in figure 2.2 is rated for operating point Q_1 and H_1 but by trimming the impeller the same pump can be used for new set point (Q_2 and H_2). The family of envelopes covers a wider range of head and flow than that of illustrated in figure 2.2. The family curves are a group of curves based on different sizes of suction, discharge, and impeller diameter in a constant speed for a particular pump. Depending on flow-capacity requirements and size of the piping several configuration might be capable to meet the system requirements. Other selection criteria that are discussed in following sections decide that which of the configurations suits the system best.

2.2.3. Power and Efficiency

Centrifugal pumps convert the received power from the shaft into kinetic energy to increase the liquid velocity and subsequently provide adequate head. True understanding of this energy transfer process and different losses helps in proper selection of pump driver and evaluation of energy efficiency of the pumping system. Pump manufacturers provide horsepower (or simply power) data of the pump through performance curves. This power is called *brake horsepower* (BHP) and refers to the actual amount of power required by pump to maintain a certain amount of head and flow. BHP or shaft power is input power to the pump or output power from the driver. It is determined by some tests by the manufacturer and is given as a function of capacity in pump performance curves for the whole range of flow handled by a particular pump.

There are some losses in the power transfer by the pump resulting in decline in power output of the pump. Output power of the pump, also called *water horsepower* (WHP), can be calculated as follows:

$$\mathrm{P}_{out} = \rho g Q H \tag{2.10}$$

P_{out} = output power of the pump (WHP) [W]
Q = flow rate [m^3/s]
H = head [Pa]
ρ = fluid density [kg/m^3]

Therefore, the total efficiency of the pump (η_{pump}) can be reduced as:

$$\eta_{pump} = \frac{output\ power}{input\ power} = \frac{\mathrm{P}_{out}}{\mathrm{P}_{shaft}} = \frac{water\ horespower}{brake\ horsepower} = \frac{WHP}{BHP} \tag{2.11}$$

The data of power and efficiency of a particular pump is usually given by pump performance curves as a function of capacity (Fig. 2.4), though the equations above can be used for direct calculation based on head and capacity over the pump operating time. There is another power related terms in pump terminology indicating the performance of pump driver called *wire-to-water horsepower*. This is the net power gained from pump and its driver to the liquid (water). If the efficiency of driver (η_{driver}, including transmission system) is known, then the net power consumption of the pumping system can be determined. For example, for an electric driver, P_{in} must be taken from available electricity source to raise the liquid head as of WHP.

$$\eta_{driver} = \frac{output\ power\ to\ pump}{input\ power\ to\ driver} = \frac{\mathrm{P}_{shaft}}{\mathrm{P}_{in}} \qquad (2.12)$$

However, it should be reminded that in case of using other auxiliary devices to transfer power (e.g. variable speed device or gear box etc), wire-to-water power (P_{in}) is determined by combination of the efficiency of all the intermediate devices replacing(η_{driver}) in equation 2.12.

2.2.3.1. Energy losses in pump

The power losses through the energy transfer from shaft to the liquid by a centrifugal pump can be summarized in four groups:

A. Hydraulic losses:

They are sum of the losses due to friction in liquid passing the impeller and volute. Losses due to continuous change of the liquid direction in the pump are also considered in this category.

B. Volumetric losses:

The leakage of liquid from the discharge part back to the suction side is a source of loss in a centrifugal pump. The leaked liquid goes through the wear rings in a closed impeller pump while through the front of vanes in open impeller ones. Having pump in operation over a long time, internal clearances are may be opened due to erosion or wear resulting in increase in volumetric losses.

C. Mechanical losses

Mechanical losses are typical frictional losses in moving parts of the machine, e.g. between bearings with seals.

D. Disk friction losses and frictional losses

Rotation of impeller (rotating disk) in close proximity to the casing (fixed disk) results in disk friction losses (P_{df}). Frictional losses in the balancing device (disc or piston) are other source of power loss (P_f).

The pump total efficiency can be explained as the combination of the efficiency of pump in dealing with these losses:

$$\eta_{pump} = \eta_{hyd} \times \eta_{vol} \times (\eta_m - \frac{\mathrm{P}_{df} + \mathrm{P}_f}{\mathrm{P}_{in}}) \qquad (2.13)$$

η_{hyd} = hydraulic efficiency [-]

η_{vol} = volumetric efficiency [-]

η_m = mechanical efficiency [-]

P_{in} = pump input power [W]

$\mathrm{P}_{df}\ and\ \mathrm{P}_f$ = disc friction and frictional losses [W]

To avoid complexity, more detailed explanation of each component of equation 2.13 can be studied in [4], [7], chapter 1, and [8]. Based on the pump configuration and complexity, some other terms can be added or combined in the terms of the equation 2.13. In pumping system design practices, it is more straightforward to divide the losses to two main parts, mechanical losses and other losses, which lead to an increase in liquid’s temperature, called *internal losses*. Therefore, the efficiency of the pump can be reduced as:

$$\eta_{pump} = \eta_m \times \eta_i \qquad (2.14)$$

η_m = mechanical efficiency

η_i = internal efficiency

The main idea of this section was to introduce the main sources of power loss in centrifugal pumps for better understanding of their performance and selection criteria.

2.2.3.1 Best efficiency point (BEP)

Pump manufacturers introduce a set of performance curves for their products to provide required data for the user. In addition to head-flow curves for different impeller diameters, pump efficiency, brake horsepower and required suction head is usually supported for a pump in a given speed. These curves for a centrifugal pump in single impeller size and fixed speed are illustrated in figure 2.3.

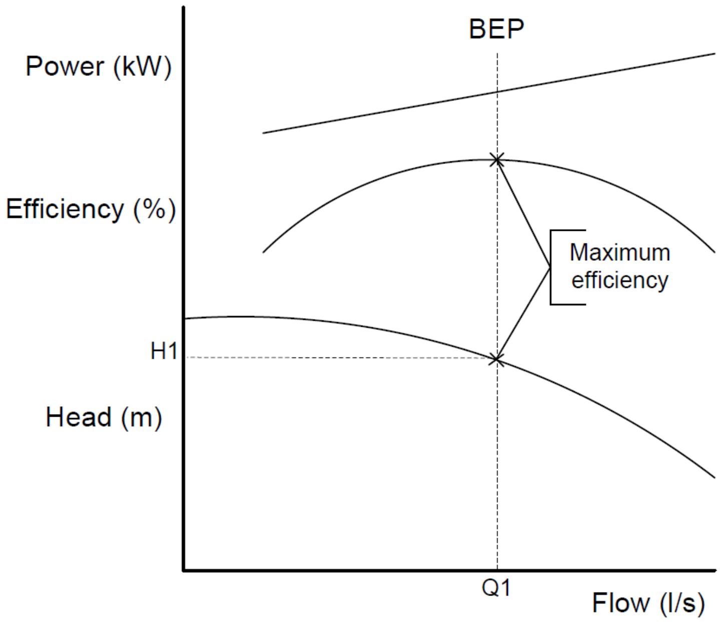


Figure 2.3 Pump performance curves for a single diameter size in constant speed

In a typical centrifugal pump, the head–flow diagram usually rises towards shutoff point developing lower amounts of flow in higher heads, and vice versa. The power curve is linearly rising as flow increases while the pump efficiency curve shows a maximum point in a parabolic behavior. Pump efficiency also varies with flow, reaching to a peak value at some intermediate point in flow regime known as the best efficiency point (BEP), then falling as flow develops. In other words, though the pump can develop higher flows or larger heads, the most energy efficient operating point of the pump remains in BEP, maintaining Q_1 and H_1.

Having the pump operating in BEP not only reduces the energy consumption based on the pumped flow, but also decreases the maintenance costs caused by excessive radial loads [7]. BEP is the working point in which radial forces acting on the periphery of the impeller are minimal. This decreases the load on the shaft and radial bearing system preventing shaft deflection or mechanical seal and bearing unexpected failure. The far operation of the pump from BEP increases the risk of the above-mentioned symptoms, typically happening in low flows operation.

It should be reminded that the pump performance curves are typically supplied in ISO format, providing data for a set of impeller diameter sizes. In this case, efficiency and power are given in constant value curves in whole operating range supported in different impeller diameter sizes. Power is usually given in values available for commercial electric drives.

2.2.4. Cavitation and NPSH

In pumping a liquid, significant effort is devoted to understand the phenomenon of cavitation and preventing its occurrence. Cavitation happens when the liquid pressure during the pumping process falls below the vapor pressure at that temperature. It means having vapor bubbles mixed with liquid flow passing through the pump. In presence of pressure increase during the impeller vane, these bubbles implode at the same point resulting in erosion of the impeller, noise, vibration, and to the extreme extent failure of the device. Cavitation diminishes the hydraulic performance of the pump that can lead to incline in head and flow, known as *break away* [6], illustrated in figure 2.4.

The excessive vibration due to cavitation results in the failure of the seal and bearings of the pump. This is the most probable failure symptom of the cavitation and one of the reasons that offer a proper understanding of net positive suction head (NPSH) and cavitation in the pumping system design and usage. NPSH is the difference between total head and liquid vapor pressure at the pump inlet, defined by DIN 24260 [7]. In a pumping system, there are two definitions related to NPSH, called NPSH required and NPSH available.

2.2.4.1. $NPSH_a$

Available NPSH is defined by the plant characteristics in which the pump is operating. It is the head introduced at the pump suction above the liquid vapor pressure. $NPSH_a$ is not dependant on the pump configuration and is calculated as follows [7]:

$$NPSH_a = \frac{P_e - P_v}{\rho \times g} + Z_e - Z_s + \frac{V_e^2}{2g} - H_{loss} \qquad (2.15)$$

$NPSH_a$ = net positive suction head available [m]

P_e = absolute pressure on the fluid surface at supply vessel [Pa]

P_v = vapor pressure of the liquid at the pumping temperature [Pa]

Z_e = elevation of the liquid in supply vessel [m]

Z_s = elevation at the suction of the pump [m]

V_e = liquid mean velocity at intake vessel [m/s]

H_{loss} = head losses in suction piping [m]

$NPSH_a$ is calculated by the pump engineer user then presented to the pump manufacturer as a part of the pump specification, besides head and flow etc. The higher the flow rate, the greater friction losses in the suction line resulting in lower NPSHa at the pump inlet, while higher head is required (NPSHr) due to more losses in the pump suction nozzle (Fig. 2.4).

2.2.4.1. NPSHr

NPSHr or net positive suction head required is the suction head needed at the impeller centerline over the vapor pressure of the liquid to avoid cavitation. NPSHr is dependent on the pump inlet design, and is not related to the suction piping system. In other words, a centrifugal pump requires a certain pressure at the suction flange higher than the vapor pressure of the liquid at pumping temperature. Liquid encounters pressure losses entering the pump, caused by frictional effects as the liquid goes through the suction nozzle, before it reaches to the impeller vane where pressure increases again. NPSHr is established by the pump manufacturer using special tests, and the value of NPSHr is then illustrated on the pump performance curve as a function of pump capacity to be considered in pump selection (Fig. 2.4).

NPSHr is sometimes shown in ISO format on the curve increasing at higher flow rates as the friction losses increase inside the pump inlet. For some pumps, NPSHr is also higher in reduced impeller diameter sizes in an unchanged flow rate. Therefore, it should be considered in impeller diameter reductions that not to fall into the cavitation area. The quantitative value of NPSH that is obtained from the curves is given based on cavitation criteria in terms of percentage of head loss. For instance, $NPSH_3$ refers to the NPSH required for the pump in the cavitation condition resulting in 3% head drop. The calculation of available NPSHa may be done in a more conservative manner considering safety margins to guarantee an adequate difference above the NPSHr for a whole operating range of the pump over its life time.

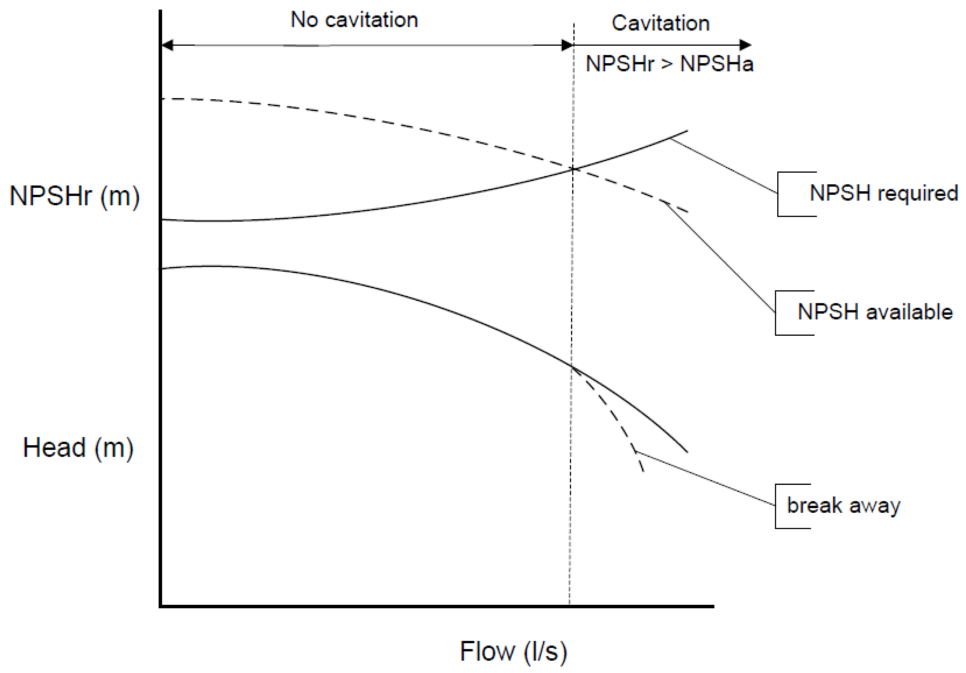


Figure 2.4 NPSH as a function of flow, and possible occurrence of cavitation

2.2.5. Specific Speed and Affinity Laws

Specific speed in pumping terminology is a dimensionless design index that is used by pump designers for describing the geometry of pump impellers and classifying their type. Specific speed for a particular pump provides information about the pump impellers shape, different performance curves for different shapes, and the variation in efficiency at the BEP for different centrifugal pumps [6]. This provides hints for the selection of best energy efficient pump for an application. Specific speed (n_q) is expressed as follows [7]:

$$n_q = N \times \frac{\sqrt{Q}}{H^{3/4}} \tag{2.16}$$

N = pump rotational speed [rpm]
Q = flow (flow per impeller eye for double-entry impellers) [m^3/s]
H = head (head per stage for multistage pumps) [m]

Specific speed is considered a dimensionless number that remains unchanged for a particular centrifugal pump through its operating range (different Q and H). It is calculated by equation 2.16, having rpm, capacity, and head at BEP picked from the performance curves at full diameter. In other words, specific speed is an indicator for categorizing different pumps without being related to head and capacity at operating point. It is also a basis for the prediction of the pump performance at different speeds (affinity laws).

Based on the specific speed, centrifugal pumps can be categorized in three main groups. *Radial flow* pumps are low specific speed pumps (below 2000), which are known for low flow and high heads. They increase the head by radial movement of liquid from the shaft. High specific speed pumps (greater than 8000), known as *axial flow* pumps, are used for developing high flows in low heads. The intermediate region between these two categories is *mixed flow* pumps that have a combination of the other two's characteristics [6]. The slope and shape of the H-Q is also related to the specific speed, showing the steepest diagram for radial flow pumps while the flattest for axial flow pumps.

Based on the constant specific speed for a pump, the affinity laws can be derived. Affinity laws are used to predict the pump's behavior in speeds other than its nominal speed or different impeller diameters. Therefore, two sets of the rules can be derived of which the laws are as follows for rotational speed change:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \tag{2.17}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \tag{2.18}$$

$$\frac{\mathrm{P}_1}{\mathrm{P}_2} = \left(\frac{N_1}{N_2}\right)^3 \tag{2.19}$$

In which, Q, N, H and P indicate pump's flow, rotational speed, head and shaft power, respectively. In second set of the laws, impeller diameter (D) can be used instead of speed (N). Equations 2.17-19 can be used for the calculation of head-capacity curves in different speeds. This is the method to predict the performance of the pump running in variable speed setting. However, it should be noted that the BEP of the pump at the new speed may change from the initial point (figure 2.5). By changing the pump's speed, the BEP efficiency does not change by value but shifting to a new head and capacity (Fig.2.5). The attention should be given to NPSH changes in varying the pump's speed.

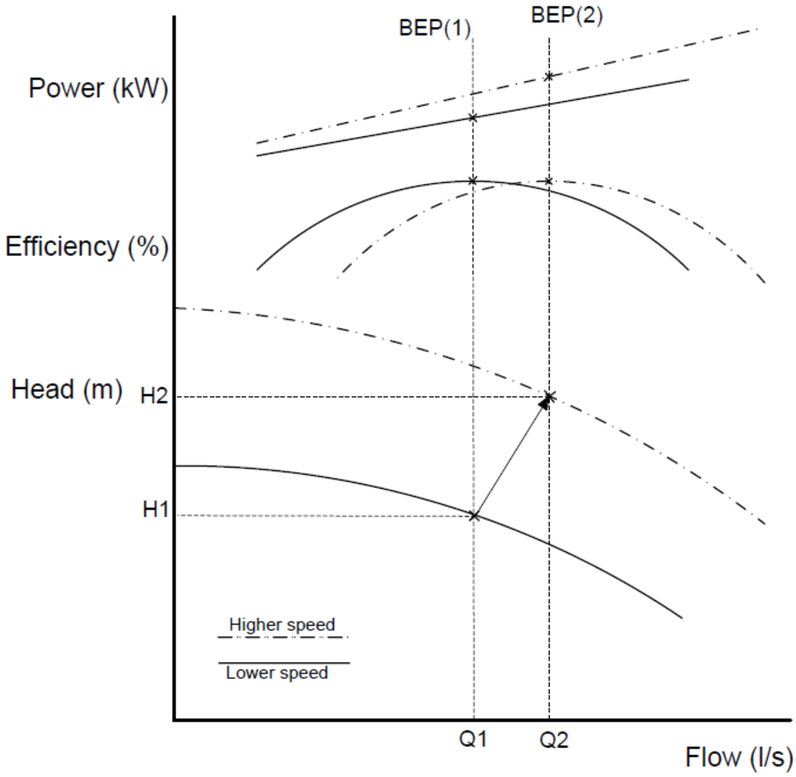


Figure 2.5 Change in performance curves and BEP point based on affinity laws

Some manufacturers offer the same relationship between NPSH and speed as expressed for head in equation 2.18, while others recommend lower powers for NPSH ratio, for example 1.5.

2.3. Centrifugal Pump Types and Configurations

There is an extensive number of configuration and mechanical features for centrifugal pumps that is beyond the scope of this study. Accordingly, it is the task of the pump user and supplier to choose the most suitable alternative for a particular application. In this section, some important definitions in pumping terminology regarding the pump’s configuration are explained, without detailed technical discussions, to provide a basic knowledge about the terms used in consecutive chapters.

2.3.1. Impeller and Casing Types

In centrifugal pumps, impellers are divided to two major groups, *open* and *closed* [6]. The way that impeller vanes are seen from the suction side is the basis for this classification. Open impeller types are primarily suitable for pumping liquids that contain solid particles. In closed impeller pumps, impeller vanes at the suction side are covered by a shroud and there is an axial-oriented hub directing liquid to the vane passes. Closed impeller pumps are suitable for relatively clean and noncorrosive liquids, high temperature pumping applications, and multi-stage pumps. In general, closed impeller pumps have higher efficiency and lower axial thrust- between 30% and 40% lower than open impeller [6].

The centrifugal pump casing can be *single volute* casing having one *cutwater* where the liquid is separated (Fig. 2.6). *Diffuser casing* is a more complicated casing including several flow passageways around the periphery of its impeller discharge. The liquid leaving impeller vanes enters the nearest flow path in the diffuser casing, rather than moving around the casing periphery in single volute casing. The main benefit of the diffuser casing configuration is the near balancing of radial forces that reduces shaft deflection and the need for a heavy-duty radial bearing system.

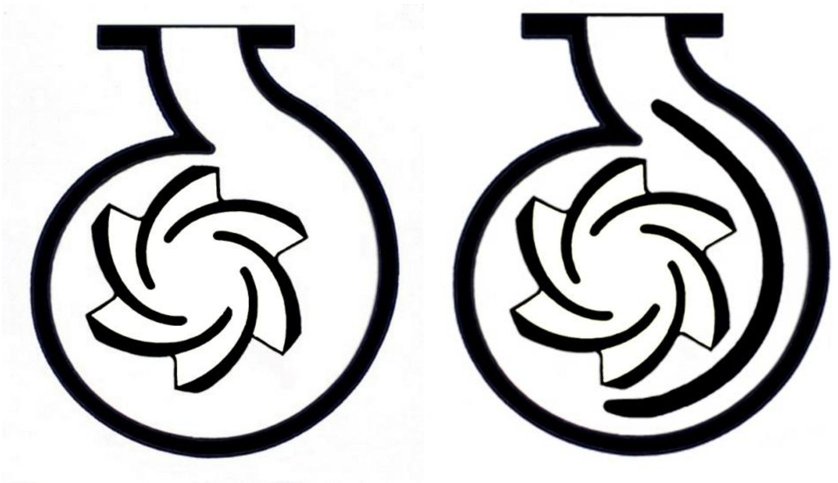


Figure 2.6 Single volute (left) and double volute (right) pump casing [9]

A combination of a single volute casing and a diffuser casing is a *double volute* casing (Fig.2.7). In double volute casing design, the volute is divided to two cutwaters resulting in lower radial loads. This casing is typically used for larger flow pumping applications allowing the use of smaller shaft and bearings. Pumps can be classified based on the assembly of their casing to axial split and radial split case pumps. In a *radial split case pump*, impellers and diffusers are joint together hold by a tube or rods. For heavier duties, like boiler feed water pumps, a special pump known as double barrel or double case is used to maintain high flow and heads.

2.3.2. Suction Methods

In terms of suction method, centrifugal pumps can be divided to single or double suction impellers. While in *single suction* impellers the liquid enters from one side of the impeller, in a *double suction* impeller liquid is received from both sides [6]. Double suction method is merely used with closed impellers and in conjunction with double volute casing, either as a single stage or the first stage of a multistage pump. The main advantage of double suction is lower, nearly zero, axial thrust as well as much lower NPSHr due to the presence of two flow paths at suction inlet. Hence, double suction impellers are more suitable for higher flows where the risk of NPSH related problems is higher.

End suction pumps are the most common type of centrifugal pumps receiving the liquid from the end and discharging it at the same side with a right angle from the shaft. *Closed-coupled* end suction pumps are directly linked to a motor eliminating pump's bearing system and common for low duty application. However, most of heavy duty, high temperature application benefit from *frame-mounted* end suction pumps, in which the motor and pump are separated but coupled with a shaft and bearing system. The latter is more suitable for industrial applications up to 1500 l/s capacity.

2.3.3. Multistage Pumps

Multistage pumps provide the highest head at the same speed by the use of multiple impellers operating in series. The liquid moves through the pump's case from each impeller to the following with an increase in head. Not only generating high heads, multistage pumps are known for higher efficiencies compared to single stage pumps providing the same duty. The design of the first stage can be different to maintain the NPSH requirements of the pump. Boiler feed water pumps are usually multistage pumps benefiting from barrel split case, closed impeller, double suction and double volute casing for relatively medium to large size applications.

3. PUMP SIZING AND ENERGY EFFICIENCY

The main performance characteristics of centrifugal pumps, including head-capacity relationship, were expressed and discussed in the previous chapter. In order to choose the most suitable pumping design for a particular application, the true understanding of the system behavior is also important. Pumps are selected to meet the requirements of the system in terms of head and capacity. The determination of system head curve is hence the first step in rating the pump. Pump selection criteria and related consideration for the fulfillment of the most energy efficient performance are other important issues that are discussed in this chapter. Variable speed pumping and multiple pumps are other topics covered in the following sections.

3.1. System Head Curve

System head curve is used to illustrate the head requirement of the system based on different flow rates [6]. System head consists of two major parts, static and dynamic components. In reference to the discussions in the previous chapter (sections 2.2.1.2-5), the static head represents the head requirements regardless the flow velocity and frictions in the piping system. Therefore, the static head, which consists of pressure and elevation head components, does not vary with flow fluctuations. *Dynamic head* or frictional head in reverse is totally a function of flow rate, starting from zero at shut-off point and increasing in higher capacities.

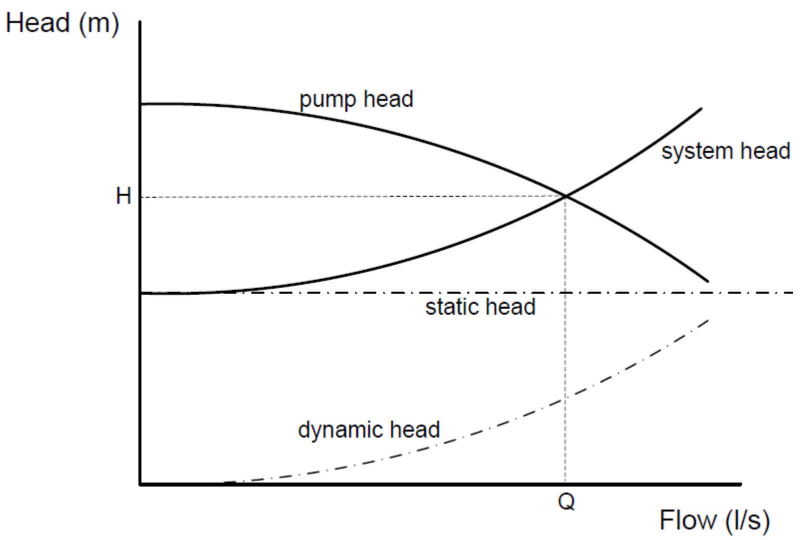


Figure 3.1 System head components and intersection with pump head curve

These two components of the system head curve and their combination is illustrated in figure 3.1.

In fact, the system head (H) is a parabolic function of capacity (Q) [10]:

$$H_{system} = H_{static} + H_{dynamic} = H_{static} + CQ^2 \tag{3.1}$$

In which, C is a constant value representing the terms producing dynamic head [10]:

$$C = \frac{f\frac{L}{d} + \sum K}{2gA^2} \tag{3.2}$$

L = piping length [m]
d = pipe diameter [m]
f = friction coefficient [-]
K = local head loss coefficient of fittings and valves [-]
g = gravitational acceleration [m/s^2]
A = pipe cross-sectional area [m^2]

Based on the equation 3.1, the higher the capacity, the more dynamic head of the system resulting in steeper system curve. Considering the system head curve and pump H-Q curve in figure 3.1, it should be noted there is one intersection between pump and system head curves occurring at design operating pump (Q, H). In case of any modification in the system resulting in higher demand for flow, the operating point moves towards the right direction on the system head curves. Having higher flow rate at new design point results in higher head demand for overcoming the extra dynamic head imposed to the system. It is obvious that the existing pump in figure 3.1 is no longer capable to meet the new head demand. However, for lower flow rates the same pump can be used while considering some solutions for decreasing the head. This will be discussed in detail in the following sections.

It should be noted that if the static head comprises the major share of the system head, e.g. boiler feed water system, the shape of the system head curve tends to flatten, with slight slope at higher flows. In this case, any minor variation in the system head can lead to great swings in the capacity.

3.2. Pump Selection

Having the system head curve behavior of a particular application, the most suitable pump can be selected for the system requirements. In fact, there may be many pumps that can satisfy the design point of the system. The right decision in selection of the pump depends on several parameters that are discussed in this section. Choosing the right and optimal pumping design not only fulfills the system requirements over a wider range of head-capacity fluctuations but also reduces the energy consumption, maintenance costs, and the risk of breakdown of the pump.

3.2.1. General Considerations

System head-capacity requirements, design operating point, off-design working regime, NPSH available at the pump inlet, and pump speed are general considerations in selection of the pump in consultation with a pump manufacturer. Providing pump specification, pump designer and manufacturer can decide about the most suitable pump configuration for a particular application. As system head and capacity are likely to change over the lifetime due to the increase in frictional losses, discharged liquid pressure and elevation difference, or flow fluctuations; the design should be so that the pump remains in an economical operation region.

Impeller trims are, to a limited extent, another way for adjusting the pump for the new head-capacity requirements. Varying the pump's speed, if applicable, is another strategy for meeting a wide range of head-capacity requirements. The combination of multiple pumps in parallel or series is another common way for heavy duty or complicated pumping systems.

3.2.2. Pump Speed Selection

The main question after rating the pump is the suitable speed for the operating conditions. As there are different alternative pumps that can fulfill the system requirements, the choice of appropriate speed depends on several criteria that come as follows. The previous experience and manufacturer recommendations are also very important in definition of pump speed.

3.2.2.1. Suction specific speed

In reference to the section 2.2.5 and equation 2.16, the suction specific speed (n_s) for a centrifugal pump can be expressed [6]:

$$n_s = N \times \frac{\sqrt{Q}}{NPSHr^{\ 3/4}} \tag{3.3}$$

N = pump rotational speed [rpm]
Q = flow (flow per impeller eye for double-entry impellers) [m^3/s]
NPSHr = net positive suction head required [m]

The values used in the equation are taken form pump BEP. The recommended value for suction specific speed (n_s) is between 7000 and 9000, more specifically 8500 [6]. After calculation of NPSHa for the system, the value of NPSHr can be determined by consideration of a safety margin (e.g. 3%). Having NPSHr and recommended suction specific speed (n_s), Eq. 3.3 can be solved for rotational speed (N). This is the maximum speed for a given pump but not the only alternative.

3.2.2.2. Pump performance curves

Based on the Eq. 2.16, pump specific speed depends on the rotational speed. As mentioned before (section 2.2.5), for a particular application the shape of performance curve is related to pump specific speed. For instance, for a radial flow pump in which the specific speed is below 2000, the performance curve best suits applications with low flow and high heads. Therefore, the pump speed should be selected in a manner that fulfills this specific speed range (Eq. 2.16).

3.2.2.3. Efficiency

Another important issue in selection of pump speed is the gained efficiency at BEP. The efficiency of a particular pump is a function of flow and pump specific speed. The higher the capacity at BEP, the greater the efficiency at a constant specific speed. Though this approach is merely theoretical, it can picture the general pattern in speed selection to gain the highest efficiency at a given design capacity.

3.2.3. Oversizing

Oversizing the pump in design and procurement process can lead to severe economic and process losses during the pump lifetime. This is typical in calculating the design point of the pump to consider some fudge factors to ensure the capability of the pump for future increase in the duty. The accumulation of solids in piping systems can, for example, result in higher frictional losses after some years. It is also a matter of fact that the pump should be able to meet the possible extra loads. Though a safety factor seems to be reasonable in these cases, the overestimation in calculation of this factor can lead to selection of oversized pump and driver. Then, to run the pump in lower flows, the flow should be throttled resulting in deviation from the BEP. It not only increases the capital cost and energy consumption of the system, but also can result in higher maintenance costs and in an extreme condition deflection of the pump. The main idea of this study is also to provide a basis for optimal design of the pump size.

3.3. Pump Control Methods

After determination of pump design point and selecting the appropriate speed and pump configuration, the pump is put into operation. However, there are several reasons that the pump should be run in a working point other than the initial design point, e.g. the need for lower flows. A typical pump can theoretically operate in a broad range of flows and heads, illustrated in head –capacity curve. Since the system curve is not following the same pattern as the pump H-Q curve, some measures should be taken to adjust the working point of the pump back to the system requirements. There are different alternatives to modify the pump output that are discussed in this section.

3.3.1. Throttling and Bypassing

In constant speed pumping, throttling the pump's discharge pressure is a typical way to conform the pump output to system requirements. If the flow rate in the system declines under any circumstances, the system head at the new operating point does decrease due to the lower frictional losses while the pump head increases simultaneously. To return back the discharge to the desired pressure at the new flow, an artificial friction loss is imposed to the system by changing the valve position. It should be noted that the system head curve has steeper slope due to the higher frictional losses in the new conditions (Fig. 3.2).

In reference to Eq. 2.10, the power consumption of the pump is directly related to the product of head and capacity, corresponding to the surface area under the pump head-capacity curve. The amount of power loss due to throttling is clearly illustrated by hatched lines in Fig. 3.1. If the system would be capable to change the pump head curve, by trimming the impeller or speed varying, the required power were only proportional to the gray area. In this example, a system rated at A_1, encounters to flow reduction to Q_2. The pump discharge head in this new flow is H_{2b} while system head has inclined to H_{2a}. To resolve the gap, system head curve is changed to intersect the pump head at B_1 by throttling the pump discharge (Fig. 3.2).

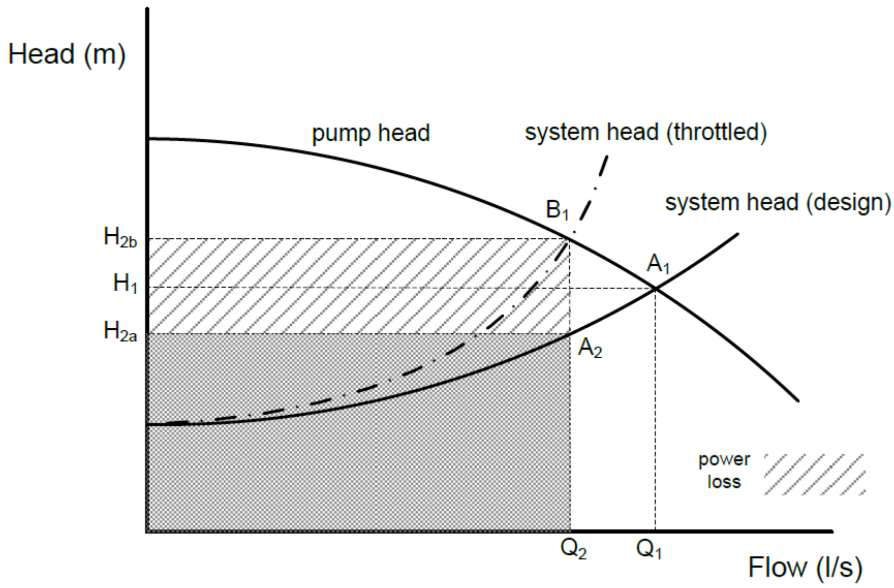


Figure 3.2 Throttling the pump discharge and corresponding power loss

Deviation of pump’s operating point from BEP is the other negative effect of throttling that can increase life cycle costs of the pumping system. Bypassing a share of discharged flow (Q1-Q2) back to the suction line is another method to maintain the new flow requirement. In this case, the pump power consumption remains constant as before while the system capacity is decreased. Bypassing is known as the least efficient way of pump control method [7].

3.3.2. On-Off Control

In some systems, flow requirements can be fulfilled by continuously switching the pump to off and on mode in a duty cycle. This method is typically applicable for those systems that include a buffer system, e.g. storage tank or drum, from which the liquid is supplied in a constant flow rate. This tolerance facilitates the pumping system for having pauses or delays in supply, resulting in off-and-on operation. As the start-up of the pump has additional losses and is not recommended in industrial applications [11], this method remains for emergency conditions for such plants.

3.3.3. Adjustment of Impeller Diameter or Changing Stage Numbers

Trimming the impeller of the pump can empower it for operation in new working condition with lower capacities. As the other parts of the pump are not exchanged or modified, impeller trimming is considered an economical solution to a specific size recommended by the manufacturer (Fig.2.2). This strategy is mostly applicable for those systems that the amount of capacity is permanently reduced and is not returned back continuously. In some circumstances, a trimmed impeller can be replaced by a new one with larger diameter if the casing of the pump can fit it [6].

In some multistage pump configurations, e.g. radially spilt casing, the number of stages can be varied for different head requirements [12]. As this solution offers shut-down and maintenance practices for the servicing pump, it cannot be considered a good solution for continuous changes in system head, while efficient for bigger alterations. It should be reminded that the impeller trimming is not usually applicable for multistage pumps.

3.3.4. Speed Control and Variable Speed Pumping

While impeller adjustment is more suitable for long-term, limited increments in flow reduction, it is sometimes required to continuously adjust the pumping head-capacity characteristics. One of the methods to meet this requirement is changing the pumping rotational speed (rpm). As mentioned before, performance of the pump in different speeds is expressed by affinity laws (Eq. 2.17-19). Similar to impeller diameter, the lower speeds shift the pump H-Q curve while keeping the trend almost similar [6] (Fig. 3.3). By reducing the pump's speed (from N_1 to N_3) for lower flows (Q_3), the produced head is subsequently lower (H_3) eliminating the need for throttling. However, it should be noted that if the pump is rated at point A_1 in BEP, it does not necessarily mean that A_2 and A_3 are also operating in BEP. Affinity curve intersects the origin of coordinates while the system head curve does not (Fig.3.3).

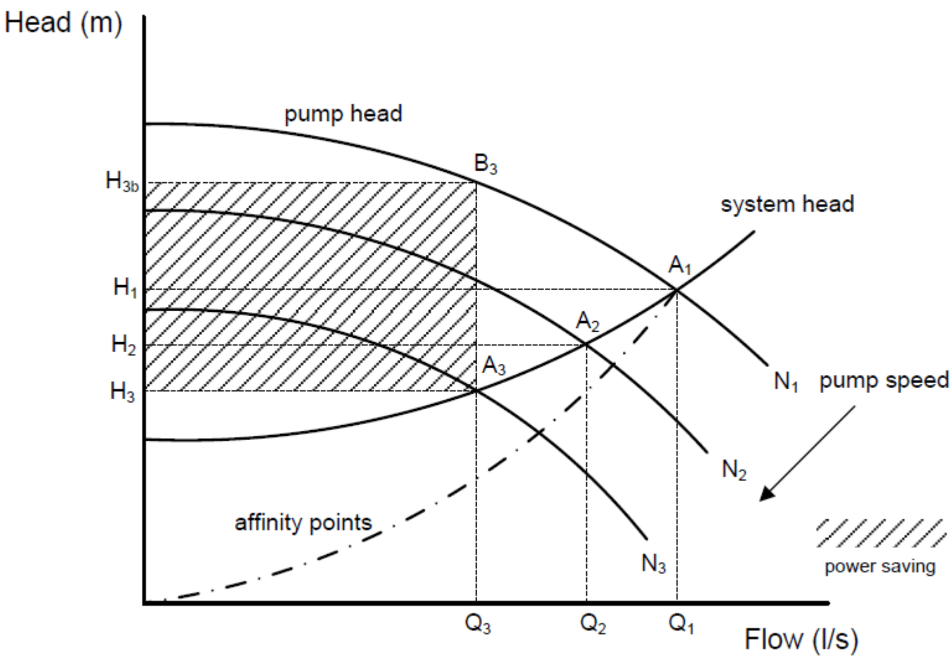


Figure 3.3 Changing the pump's speed and associated power saving

In variable speed pumping, the speed of the pump is continuously adjusted to the system requirements for lower or higher flows. There are different methods for varying the pump's speed. The speed can be changed directly from the pump's driver system, or by using a transmission system for changing the constant-speed from driver. These are discussed further in the following sections.

3.4. Multiple Pump Systems

Combination of two or more pumps can be used to fulfill the system head-capacity requirements. Multiple pumping systems allow reaching higher flows and heads using smaller units with more flexibility. There are two modes of configuration, series and parallel, which are discussed in this section.

3.4.1. Parallel Operation

Utilization of pumps in parallel increases the flexibility of the system to meet a wider range of flow fluctuations. In principle, pumps in parallel extend the pumping curve further to the right [6]. This fact is illustrated in Fig. 3.4 for two identical parallel pumps. Using two identical pumps in parallel doubles the flow for certain heads. However, it does not mean that if one pump supplies the flow of Q1, using two of them in parallel produces double flow (Fig. 3.4).

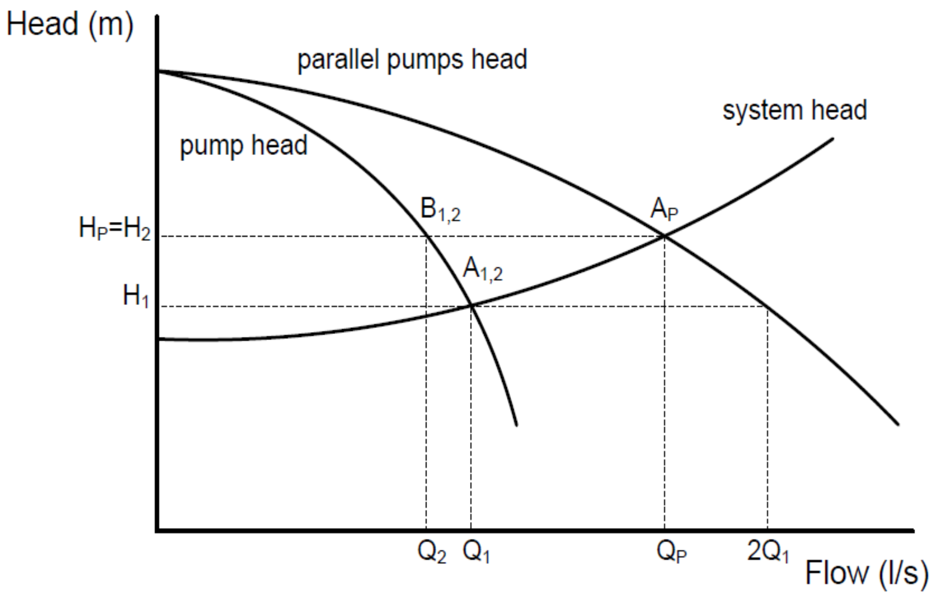


Figure 3.4 Parallel pumping system (two identical units)

Since the system head curve is not flat, it intersects the parallel pumps head curve at Ap which has a higher head compared to A1. Consequently, two pumps should be run at points B1 and B2 to produce the desired head (Hp). This example shows that bringing a new pump into the system in parallel changes the behavior of the initial pump, causing backward movement in pump head curve (from A1 to B1). This fact should be carefully considered to avoid operating pumps in unhealthy points after parallel combination.

3.4.2. Operation in Series

Pumping in series is a favorable solution for incremental increase in the head of the system, e.g. in pipeline applications. Pumps in series can produce high heads that are not economically possible to reach by single units while maintaining the desired flow rate. In power plants, a good example of this combination is the use of a booster pump to produce an initial head (NPSHa) for the following pump to prevent cavitation [13]. Using the pumps in series also diminishes the need for the use of high-pressure intermediate equipment in piping system resulting in lower costs and complexity. A combination of two identical pumps in series is illustrated in Fig. 3.5. If two single pumps operate at point A1 and A2, the combination of them in series produces the upper head curve intersecting the system head curve at As. Hs is the developed head by two pumps that could not be achieved by the use of a single pump.

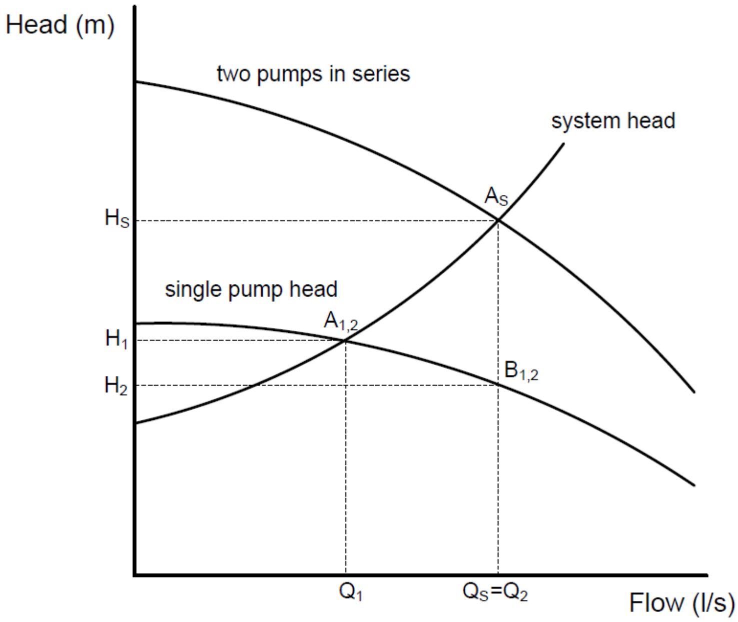


Figure 3.5 Series operation of two identical pumps

It should be noted that the new operating point of two pumps has moved farther in their head curve to B1 and B2, resulting in lower head (H2) for each pump. The other obvious fact is the total flow is also increased through the operation in series (Fig. 3.5).

3.5. Pump Drivers and Transmission Systems

In order to rotate the shaft of a centrifugal pump a driver is needed. There are different sources for driving the pump including mechanical force, steam, and electric drivers. The choice of driver is highly dependent on the size of the system, economical features, and available alternatives.

3.5.1. Mechanical Drives and Heat Engines

In principle, any device capable for providing continuous rotational power can be considered as a pump driver. Compressed air engines (pneumatic engines), internal combustion engines, and gas engines are some examples that can be used in pumping applications [7]. For instance, having a diesel engine stand-by next to the electric drives can be considered a good alternative for systems with the need of high reliability. The ability for providing a wide range of speeds is one of the advantages of these engines. Steam turbines are other example that is discussed separately as they have been more frequently used in power plant applications.

3.5.2. Steam Driven Pumps

A steam turbine can be used for driving the pump's shaft by steam expansion. Since steam is directly used for rotating the pump, some of conversion losses are diminished. This practice is, for example, used in high capacity BFPs in large supercritical plants. The shaft can be whether directly connected to the generator shaft or separately installed. The required steam and the return, in forms of condensate or back pressure, can be chosen from different configurations based on plant layout [2].

3.5.3. Electric Drives

Electric drives are by far the most frequent pump drives for a wide range of applications, from small to ultra large pumps. Pump's shaft can be directly connected to the motor or a transmission system can be used for speed variation, e.g. a hydraulic system. The electric motors can be one of the asynchronous (squirrel-cage and slip-ring induction motors) or synchronous motors [7]. The electric motor can whether run at constant or variable speed.

3.5.4. Driver's Speed Control Methods

Controlling speed of the pump drive, and subsequently the pump itself, can reduce the energy and maintenance losses due to throttling. For those drives that are inherently speed controllable, e.g. steam turbine drives or gas engines, the pump' speed can be directly regulated by controlling the driver. For example, regulation of the amount of steam directed to the turbine connected to pump can change the pump's speed.

In electric motors, the speed of squirrel-case asynchronous motors can be controlled by frequency or voltage change for continuous variation, and *pole switching* for step changes. For the other asynchronous motor type (slip-ring induction), the speed control can be achieved means of resistance in the rotor current circuit (slip losses) or by means of *sub-synchronous converter cascade*. Although synchronous motors are designed for constant speed, in some cases their speed can be controlled by static frequency converters.

3.5.5. Coupling Methods

Centrifugal pumps can be connected to the electric motors by means of a transmission system for speed regulation. Variable speed gearing between pump and motor can be one of the following methods:

- Mechanical speed converters (step-less): belt drives and friction wheel drives
- Hydraulic speed converters: hydrostatic drives and hydrodynamic converters
- Electro-magnetic speed converters

The choice of suitable coupling method depends on different parameters, among which price, rated capacity of the motor, required space etc. Some studies have shown, in a similar load, the efficiency of the variable frequency drive (VFD) is higher than hydraulic coupling, 97.3 % compared to 93.7%, for 100% speed (full load) [14]. For lower loads, VFD shows even much greater efficiency compared to hydraulic coupling, 97% and 87%, respectively. The more detailed examination of different variable speed technologies is beyond the scope of this study.

3.6 Energy Efficiency Considerations

Pumping applications are accounted as one of the high energy consumer auxiliary units in industrial plants with great potential for energy saving [15]. Having the pump operating near the BEP is one of the concerns to reduce the amount of energy consumption and other costs arisen from unhealthy operation far from BEP on the curve, e.g. low flow or run out. Optimized rating of the pump and driver for the whole service flow spectrum should be carefully studied considering the entire life cycle costs.

3.6.1. Pump Life Cycle Costs

Life cycle cost (LCC) components of the pumping system should be evaluated to determine the most efficient pumping system for a certain application. The main components of LCC analysis are initial costs, installation and start-up costs, energy costs, operation costs, maintenance costs, downtime and production loss costs, environmental costs, and decommissioning/disposal costs [16]. The main LCC components in a typical industrial pumping system are illustrated in Fig. 3.6 [17]. Energy consumption costs are the highest share for power plant applications.

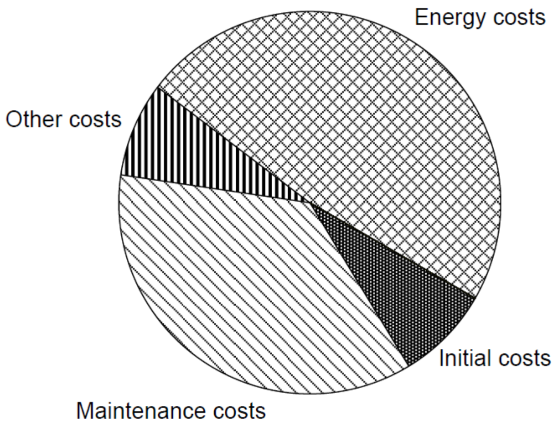


Figure 3.6 Life cycle costs for a typical medium-sized industrial pump

3.6.2. Energy Efficient Design of Pumping System

True understanding of the system requirement and pumping system characteristics is essential for an optimized pumping system design. There are several considerations to ensure an energy efficient pumping system. First, the design flow should not be oversized by overestimated losses or unnecessary safety factors. Oversizing the pump results in operation of the pump in flows lower than BEP resulting in acceleration in wear-out and higher energy consumption. The accurate selection of right pump configuration, right number of stages, suitable pump speed, and driver size for whole the flow range should be carefully studied to eliminate the need for extra costs. It should be reminded that the choice of most energy efficient alternative does not necessarily guarantee the lowest LCCs. The use of variable speed drive (VSD) is also considered to be an efficient option to avoid throttling losses.

3.6.3. Variable Speed and Energy Efficiency

Variable speed pumping (VSP) is gaining more popularity due to improvements in the technology of controlling methods. VSP can be a cost efficient choice for application with highly varying flow, including power station pumps. The energy saving opportunities in VSP was discussed in section 3.3.4 and was illustrated in Fig.3.3. The use of VSDs not only cuts energy costs but also keeps the operating point at BEP. However, initial and maintenance costs of such systems besides other constraints, e.g. space limit or the need for air-conditioned environment, should also be considered in design process.

4. BOILER FEED WATER PUMPING SYSTEM

Steam boilers are used in different process industries and power stations to produce high pressure superheated steam. BFP is a vital component that must withstand continuous operation of the plant as well as transient conditions in load. Although BFPs are available in a wide range of sizes and configurations, the most common BFP used in central power stations are diffuser/volute, horizontal, double-case barrel, single and double suction first-stage impeller, multistage, centrifugal pumps (Fig. 4.1).

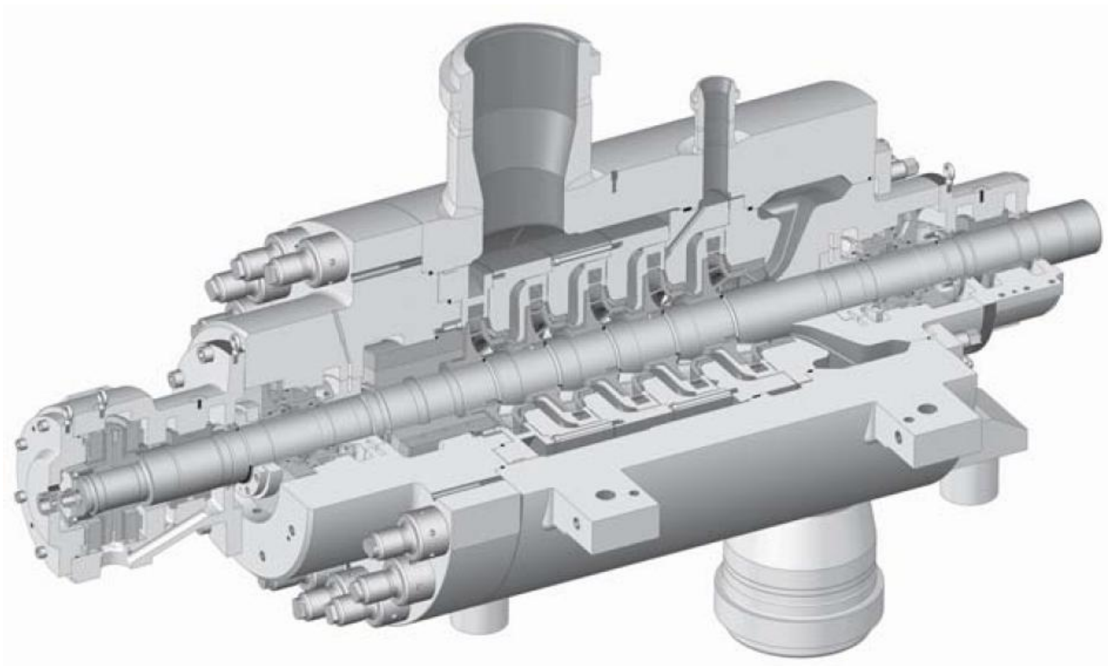


Figure 4.1 Radially-split barrel-casing multistage boiler feed pump [18]

4.1. System Characteristics

In order to ensure a proper design for BFPS, a true understanding of the system targeted for these pumps is important. BFPs are known for high demand of reliability and availability. BFP is a part of the boiler feed water system. A general boiler feed water system includes a deaerator at some elevation above the BFP suction level to provide a reservoir of deaerated, heated condensate water, as well as adequate NPSHa for main BFP [13]. The suction pipeline is designed so that limits the friction losses and maintains the required flow velocity to the BFP. A booster pump may be included before the main BFP to provide the required suction head. The booster pump's driver can be separated or its shaft can be connected to the main BFP's driver.

Discharge of the BFP includes recirculation stream back to the deaerator. There are also control valves to monitor the system flow requirements. High pressure feed water heaters (HP-FWH) are other components of the boiler feed water system that are installed after BFP. A typical boiler feed water system including BFPs is illustrated in Fig. 4.2.

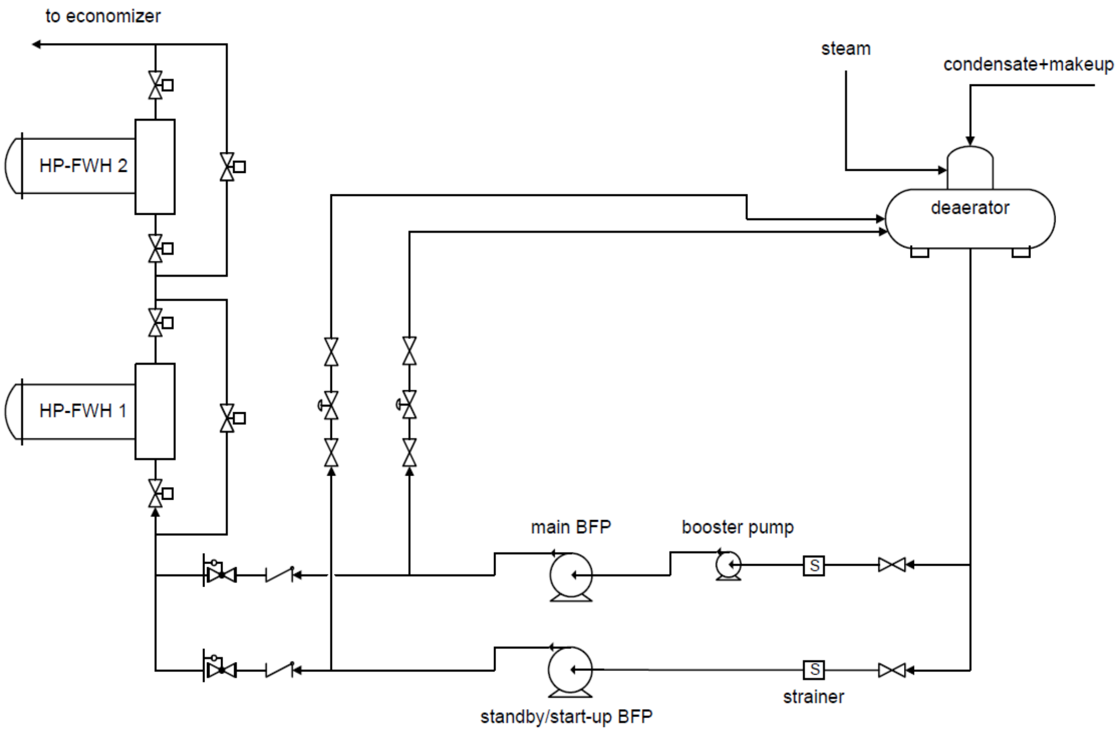


Figure 4.2 Typical boiler feed water system (booster pump + main BFP + standby)

4.1.1. Design Point

BFPs are characterized for high static head and capacity. The system head curve tends to be flat over a wide range of flow rate since the frictional losses are small compared to the boiler pressure. Flat-shape system head curve can be potentially a proper reason for the use of parallel pumps. The amount of design capacity of a BFP is calculated as follows [13]:

$$Q_{BFP} = Q_{steam} + Q_{bd} + Q_{sb} + Q_{aux} + Q_{desup} \qquad (4.1)$$

Q_{BFP} = BFP design capacity

Q_{steam} = turbine throttle steam flow (open valves and typically 5% overpressure)

Q_{bd} = steam cycle blowdown/make-up flow

Q_{sb} = soot blowing steam flow (if applicable)

Q_{aux} = auxiliary or process steam flow (if applicable)

Q_{desup} = reheat steam desuperheater flow (if applicable)

A fudge margin of 5% is typical for making the system capable for future load rises [13].

The total design head developed by BFP is determined by difference between total discharge head and total suction head:

$$H_{BFP} = H_{dis} - H_{suc} \tag{4.2}$$

H_{BFP} = BFP total developed head [m]

H_{dis} = total discharge head [m] (calculated by Eq. 4.4 below)

H_{suc} = total suction head [m] (calculated by Eq. 4.3 below)

The total suction head of the system can be calculated as follows:

$$H_{suc} = P_{deaerator} + H_{static} - H_{friction} \tag{4.3}$$

$H_{deaerator}$ = deaerator pressure (at maximum turbine heat balance) [m]

H_{static} = static (elevation) head between deaerator low water level and BFP centerline [m]

$H_{friction}$ = frictional losses between deaerator and BFP suction connection [m]

As discussed earlier, the total suction head is also important in BFP hydraulics considerations to prevent cavitation by maintaining the adequate NPSHa for the pump. In other words, a suitable pump should be selected that its NPSHr would be amply lower than the system NPSHa. Industrial practices have proved that a safety margin of 50% above the manufacturer NPSHr (at 3% head reduction) is usually sufficient. Total discharge head (H_{dis}) is also calculated as follows:

$$H_{dis} = P_{turbine} + H_{tot,st} + H_{tot,f} \tag{4.4}$$

$P_{turbine}$ = turbine throttle pressure (at maximum turbine heat balance) [m]

$H_{tot,st}$ = total static (elevation) head between BFP and drum water level [m]

$H_{tot,f}$ = total frictional losses between BFP discharge and turbine [m]

The last term in Eq. 4.4 include the losses in piping between BFP and economizer, economizer friction loss, superheater friction loss, and main steam line friction losses. After determination of the total developed head by BFP (Eq. 4.2), a design safety margin of 5% is added to determine the design total head developed by BFP [13]. A schematic of the BFPS including head and NPSHa elements are illustrated in Fig. 4.3.

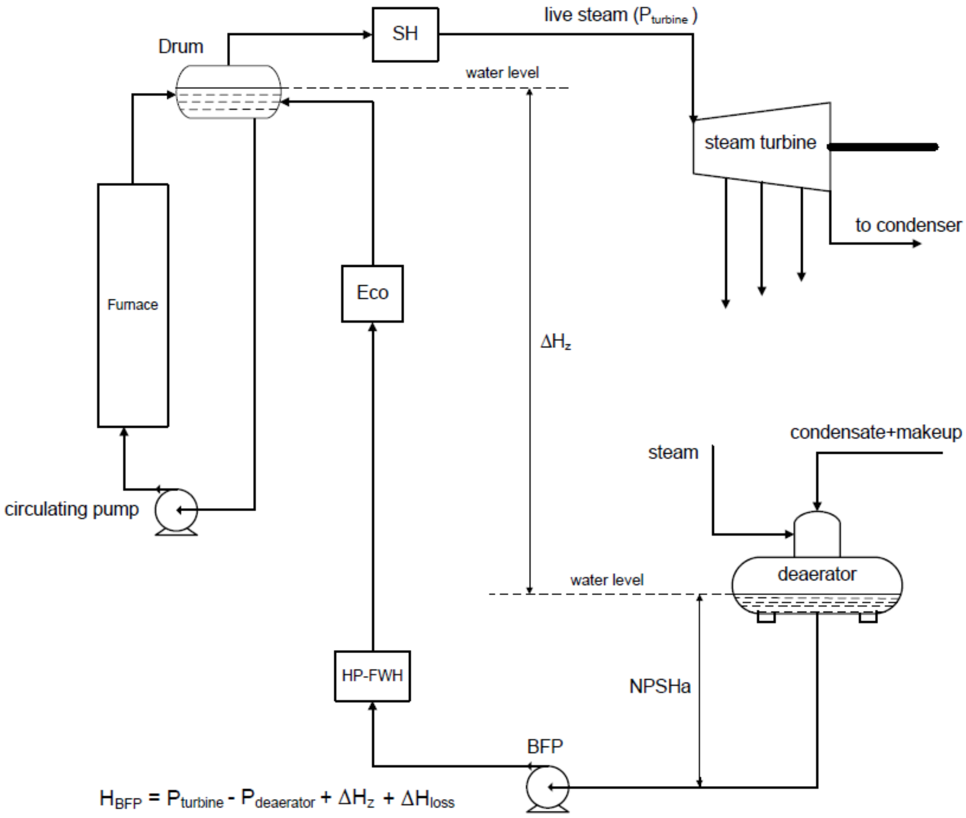


Figure 4.3 Head components of the boiler feed water pump (BFP)

4.1.2. Availability and Reliability

The continuous and steady performance of the BFP, as the heart of the boiler plant, is important in reduction of production losses. To avoid load interruption and drooping in pump head curve, BFPs are designed with low specific speed (1000 to 1800) [13]. Low value of specific speed ensures uniform, smooth, and stable head-capacity characteristics that is important in variable load operation. The continuously increasing head-capacity curve implies that BFP can be used in parallel from minimum flow to runout without problems such as excessive vibration, pressure pulsation, and cavitation. Head per stage is also another parameter in ensuring the reliability of the BFP. In general, head per stages higher than 670 meters are not recommended [13].

From system design viewpoint, the combination of BFPs in parallel and series can improve the reliability of the BFPS. A booster pump prior to the main BFP can ensure the required NPSH as well as reduction of suction recirculation in main BFP. Stand-by BFP with different driving power is another typical practice for increasing the availability of the system.

4.1.3. Suitable Driver for BFP

Suitable pump drive can be determined based on the size of the plant and available alternatives. Electrical motor drives are typical pump drives for small and medium sized boiler plants. They can be directly connected to the pump shaft or be run by means of variable speed coupling devices. Steam driven pumps are usually used in large power plants. There are diverse steam sources for driving the pump in a large power station (higher than 300-400 MWth) with different treatment of the outgoing stream [2]. A reserve pump with different power source for start-up and emergency cases is recommended for steam-driven pumps.

4.1.4. BFP Standard Requirements

Boiler standards introduce other head and capacity requirements that should be taken into account in selection of pumping systems. European Committee for Standardization has a set of technical standards (EN). Standard EN 12952-7 deals with *Water-tube boilers and auxiliary installations: Requirements for equipment for the boiler*. In addition to the requirements for head and capacity, there are instructions for the use of multiple pumps in the BFPS.

4.1.4.1. EN flow requirements

In section 5.1.1 of this standard, the requirements for the number of BFPs are described. Based on the sections 5.1.1.1 and 5.1.1.2, the safety measures would be enough to cut off the heat supply and relieve the accumulated heat in boiler, a single pump can be installed for the duty, otherwise at least two BFPs. In section 5.1.2 of EN 12952-7, capacity of the BFP, the standard requires [19]: “*The feed pump capacity shall correspond at least to 1.25 times the allowable steam output of all* *steam boilers. For safety reasons 1.15 times of maximum continuous rating is enough*". If the amount of blowdown exceeds 5% of the steam output, this amount should be also added to the maximum capacity.

4.1.4.2. EN head requirements

For head requirements of BFP, the EN standard demands the flow quantities corresponding to 1.1 times the allowable steam output pressure, in addition to the previous flow requirements. Based on the section 4.1.1 of this study and EN 12952-7 standard, the head-capacity requirements for the BFP can be illustrated as Fig. 4.4. These are the working points that must be covered when selecting a BFP. However, there is no obligation to meet both head and capacity requirements of the EN standard at one single point [20]. Fig. 4.4 illustrates the selection process of BFP to meet all the duty requirements. It also reveals the imposed power loss to the system from the first day of operation, if constant speed BFP is used. This is partly because of conservativeness in engineering calculation (oversizing), safety margins to meet the possible increase of load in the future, and standard requirements.

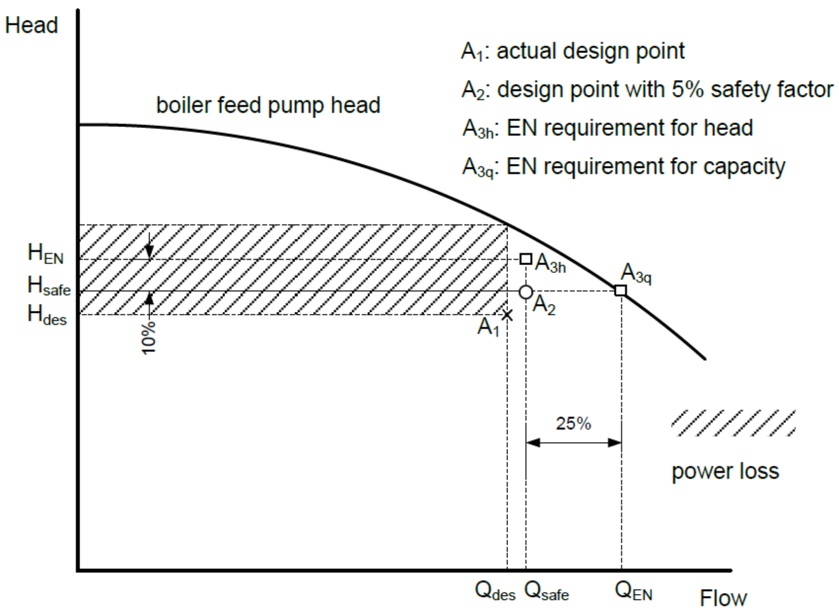


Figure 4.4 Selection of the single BFP with safety margins and requirements of EN standard

4.2. Boiler Feed water Flow and Pressure Control in Part Load

Controlling strategy of the output pressure and capacity of the boiler sets the BFP head and flow control method. The live steam mass flow outgoing form the boiler is decides the output of the plant. For steam power plants, the output load depends on live steam mass flow, temperature and pressure. In load transient conditions, the live steam temperature is usually kept constant to gain better efficiency and to avoid thermal stresses in the turbine blades [21]. Therefore, the turbine pressure is the other designating factor to control the output load. There are two main methods for controlling the turbine pressure, constant pressure operation and sliding pressure control. The main controlling streams and units for maintaining the desired flow and head in BFPS are illustrated in Fig. 4.5.

4.2.1. Constant Pressure Operation

In this method, the output pressure of the boiler remains constant during the load variation. The output load of the turbine is then adjusted by whether throttling valve before turbine inlet or by governing control of the turbine inlet nozzles. Despite marginal pressure change delays in drum boilers, this method results in constant pressure at the boiler and BFP during the load changes. In other words, to control the pumping system in part load operation, the only changing parameter is the pump capacity.

4.2.2. Sliding Pressure Control

In this method, the boiler output pressure is also varying during the load fluctuations. Using this approach in part load operation, the pressure reduction in the turbine is maintained by reduction of the live steam pressure. Though this method diminishes the throttling losses and reduces the BFP power consumption, the temperature changes in evaporator due to the pressure variations is complex to control [21]. In general, each of these controlling methods can be used for any boiler type. In practice, however, the controlling method depends on the boiler type. In the following section, the difference between two major boiler types is briefly explained.

4.2.3. Effect of the Boiler Type on Pumping System

There are two major boiler types in the industrial scale, drum boilers and once-through boilers. In addition to pressure control method which discussed above, the boiler type and associated features are important in optimizing the BFP for the whole load spectrum.

4.2.3.1. BFP control in drum boilers

In drum boilers, a storage capacity provides live steam resulting in some delays in flow demand. The changes in water level in drum transfers to the control valve so that the flow can be controlled by pump (Fig. 4.5). Live steam properties in drum boilers are usually controlled using the constant pressure method. Therefore, the total head developed by the BFP is approximately constant during the load variations. It results in a flat-shape system head curve that is amply dependant on static head raised from boiler pressure demand. Since medium-sized boiler plants are typically equipped by drum boilers, this type of boiler and relevant BFP control practices are examined in this study.

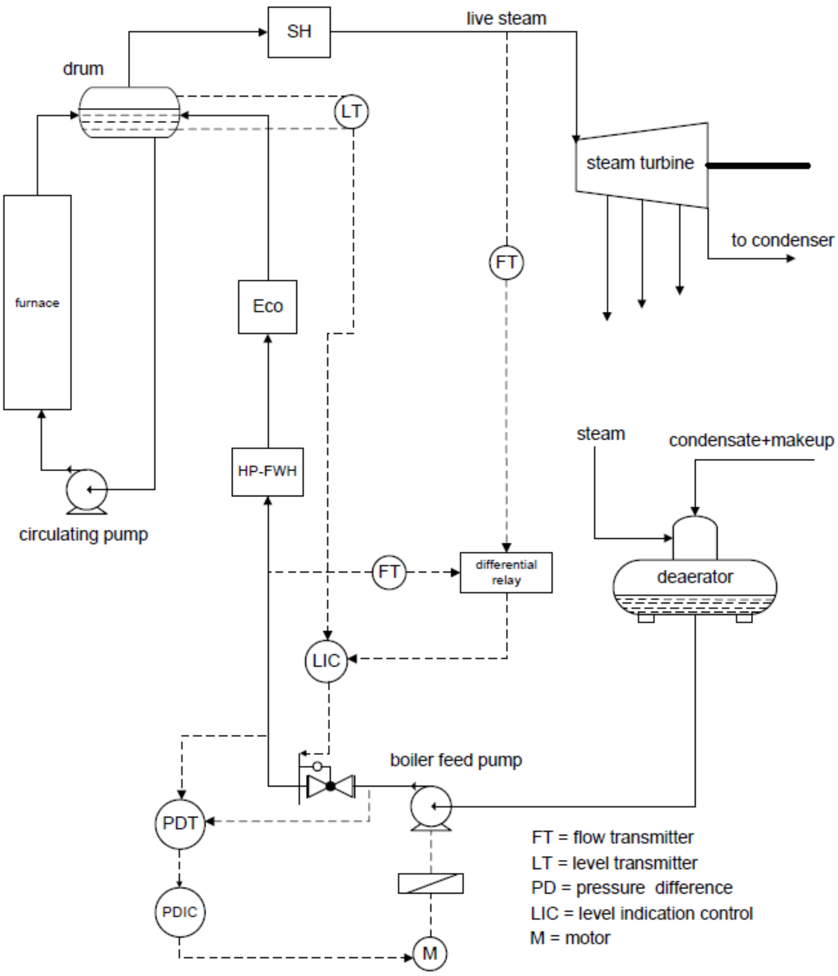


Figure 4.5 Flow and pressure control system in a drum boiler

4.2.3.2. BFP control in once-through boilers

The flow demand of once-through boilers directly decides the BFP capacity. Since there is no storage system in this type of boilers BFP must respond very quickly to load changes. Another important feature of once-through boilers is the sliding pressure control method that changes the BFP’s head consequently. As these boilers are mostly used in large plants, their BFP systems are not investigated in detailed in this thesis.

4.3. Optimal Design

BFPs are among the highest energy consumers in boiler auxiliaries subject to increase the net energy efficiency of the plant with optimal design. Optimal design of the BFP can relate to three steps. The first step relates to increasing the hydraulic performance of the pump through enhancement in pump configuration, manufacturing design, material improvement, fluid dynamics studies etc that target the construction stages. The optimal selection BFP, driver, and VSP method for the operating conditions, as well as piping considerations are the second step in reducing losses in the BFP. Precise engineering practices and good consultation provided by the manufacturer can contribute to this goal, e.g. be elimination of oversizing. Finally, performing system analysis, design, and engineering is another important strategy in increasing the energy efficiency of BFPS. The focus of this study is mainly on this segment but addressing the second approach as well.

4.3.1. System Based Approach

System level engineering is one important research area in improvement of energy efficiency of the industrial plants. It comprises system analysis, integration and optimization instead of merely focusing on individual components [22]. For BFPSs, the system approach involves the optimized design of the BFPs (single or multiple), driver, and speed control strategy for the whole operating range of the system during the life cycle. The optimization goal can be minimization of the LCC or energy consumption. Since BFPs are traditionally oversized in favor of availability, the new advancements in pump, driver, and speed control technologies increase the reliability of the optimization practices.

5. OPTIMAL DESIGN OF BOILER FEED WATER PUMPS

In the empirical part of this study, an optimal BFPS is designed for a real power plant, based on available annual load data. Based on the literature provided in the previous chapters, the optimality of the design is examined based on different criteria, like life cycle costs and energy consumption. Since the choice of energy efficient BFPS is one of the practical problems in boiler plants, the method, results and conclusions of this study can be used for other similar instances.

5.1. Input Data

In this section, the required input data of the optimization problem is examined. To find the optimal solution for each problem setting, the method by which the data is obtained or calculated is discussed in details. The efforts have dedicated to provide a set of reliable input data to increase the accuracy and applicability of the results.

5.1.1 Experimental Data of Flow

As discussed earlier, one of the bases of this study relates to the fluctuations in water capacity of the BFPs. Not only the initial design point is important in the selection of an optimized BFPS, the load pattern during the pump operation can affect the selection process. To start the mathematical modeling and definition of the optimization problem, a real annual load data of a boiler plant operating in Finland is received and analyzed. The load pattern is presented in Fig. 5.1 for one year.

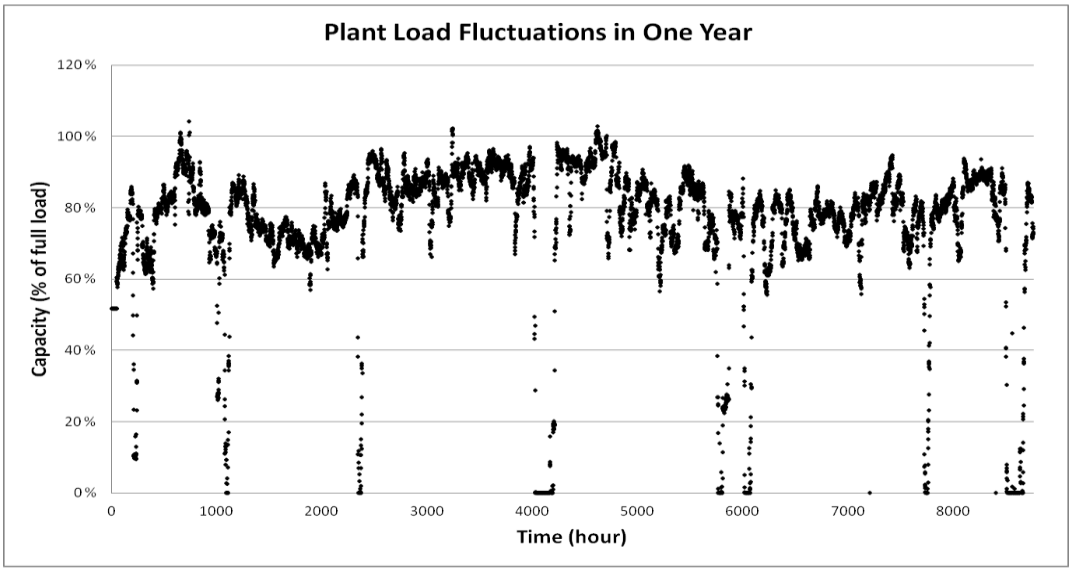
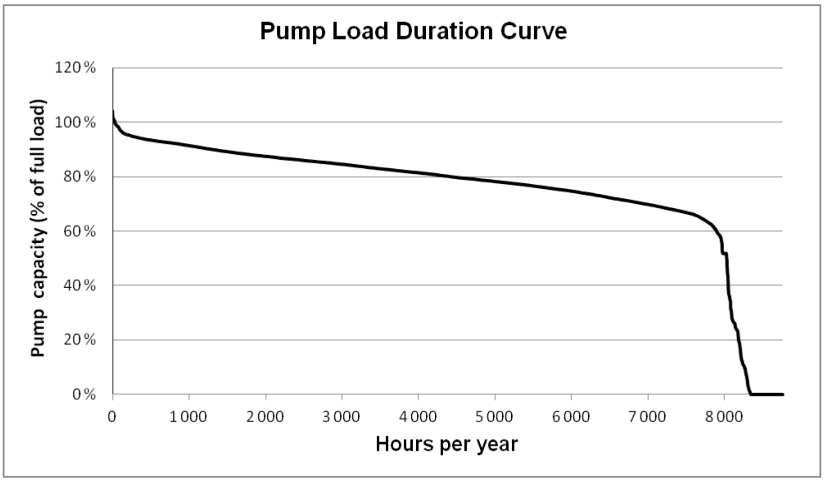


Figure 5.1 Annual load variation in a boiler plant

The data is then sorted and portrayed in terms of annual load duration curve and load duty cycle (Fig. 5.2). Based on the data, the change in water capacity of the system is fairly distributed over time, providing a good example for variable load pumping.



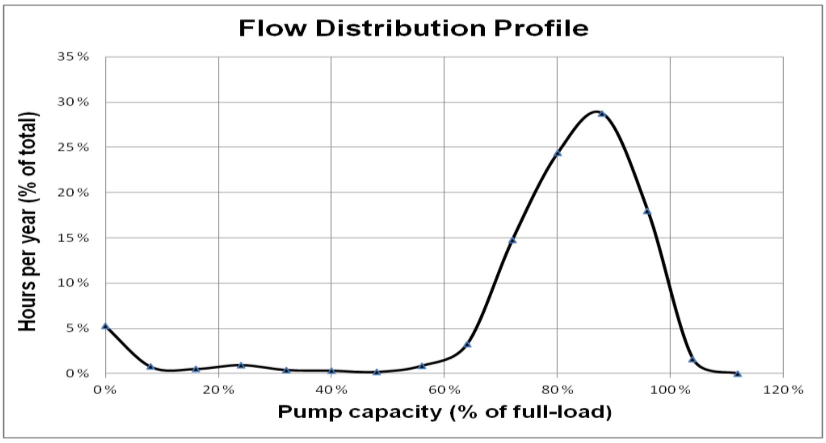


Figure 5.2 Load duration curve (above) and load duty cycle (below) of the system

It order to produce the load probability distribution profile, the flow measurements are rounded by a domain of 10 lit/s. It means in Fig. 5.2 below, the marker points represent a flow range of ±5 lit/s around the point. While it is difficult to predict the load pattern before the operation of a new plant, it is possible and beneficial to use the real load data in the substitution of old BFPs or in retrofitting a boiler plant for the improvement of energy efficiency.

5.1.2. Pump Alternatives

One of the basic requirements for making an optimal selection for BFPS is to have a set of feasible alternatives. The BFP type (based on the configuration) is one of the parameters that must be considered in optimization of BFPS. In real life engineering problems, this process resembles the selection of the BFP among a set of alternatives proposed by one or several manufacturers. The results can be however used for making more sophisticated specification for the optimal choice in design and engineering steps to find the best fit in the market.

It was important in this study to provide a diverse, practical, and reliable set of pumps for comparison and analysis in different problem settings. To ensure practical and usable results, the input pump data should be as realistic as possible. The data of the pumps used for the analysis in this study are for a set of pumps manufactured by Goulds Pumps Inc. First, by reviewing the products features, the most suitable pumps for BFPS are chosen from models 3360, 3393, and 3311 [23]. Then, the required performance data is acquired from pump selection tools provided by the manufacturer. This ensures the reliability and applicability of the technical data. A sample of selection process and data acquisition is portrayed in figure 5.3.

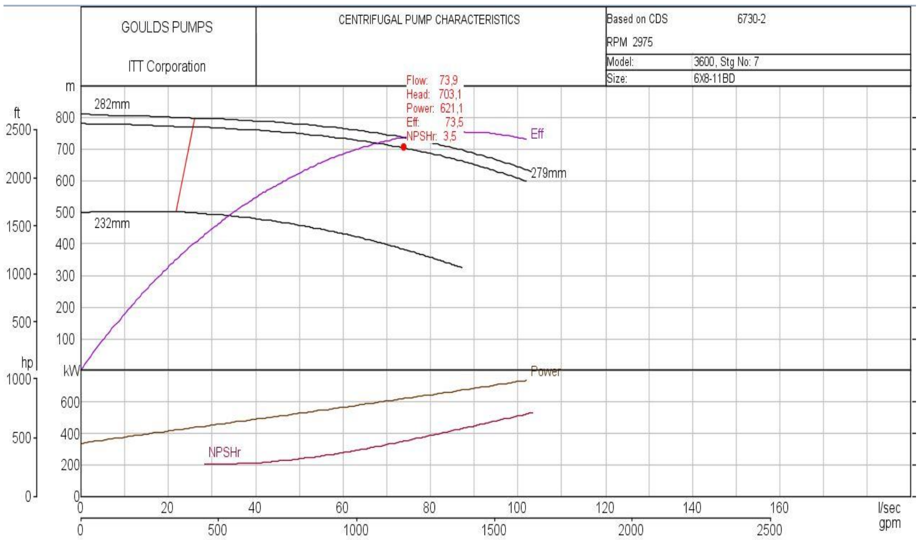


Figure 5.3 Pump data for a tentative operating point (Goulds Pumps ITT tool)

Then, the data of the performance curves for one stage of the pumps is summarized, and the trends of head and power are derived. To ease the referencing to the products, the main models used in this study and their main characteristics are collected in table 5.1. These coefficients presented in table below are used in head (H_{pump}) and power (P_{pump}) curves based on equations below:

$$H_{pump} = aQ^2 + bQ + c \tag{5.1}$$

$$P_{pump} = \mathrm{a}'Q^2 + \mathrm{b}'Q + \mathrm{c}' \tag{5.2}$$

It should be noted that power curves could be approximated with linear functions, but polynomial is more accurate [24].

Table 5.1 Pump models and their head and power curves

	Data from manufacturer [23]		Head equation coefficients			Power equation coefficients		
One stage	Model	pump ID	c	b	a	c'	b'	a'
			meter	m/(lit/s)	m/(lit/s)^2	kW	kW/ (lit/s)	kW/ (lit/s)^2
pump1	3600	6x8-11B	112	0.115	-0.002	43.6	1.124	-0.002
pump2	3600	6x8-14AD	94	0.134	-0.005	32.5	0.858	-0.002
pump3	3600	6x8-14AD	71	0.083	-0.002	33.2	0.500	-0.001
pump4	3600	6x8-13DX	58	0.079	-0.002	20.9	0.519	-0.002
pump5	3600	6x8-14AD	193	0.172	-0.003	81.7	1.875	-0.004
pump6	3600	6x8-11BD	159	0.191	-0.004	49.2	1.850	-0.004
pump7	3600	4x6-11B	106	0.265	-0.009	12.6	1.396	-0.007
pump8	3600	4x6-11A	90	0.138	-0.008	12.4	0.975	-0.004
pump9	3600	4x6-10D	70	0.324	-0.011	15.1	0.605	-0.003
pump10	3311	4x5-11C	60	0.221	-0.010	8.4	0.675	-0.004
pump11	3600	4x6-10D	173	0.172	-0.010	23.1	2.223	-0.009
Pump12	3600	4x6-11BD	145	0.228	-0.012	19.7	1.674	-0.006

The performance curve of the different pumps is portrayed in Fig. 5.4 for one stage. The more detail of the performance of each pump will be discussed where needed.

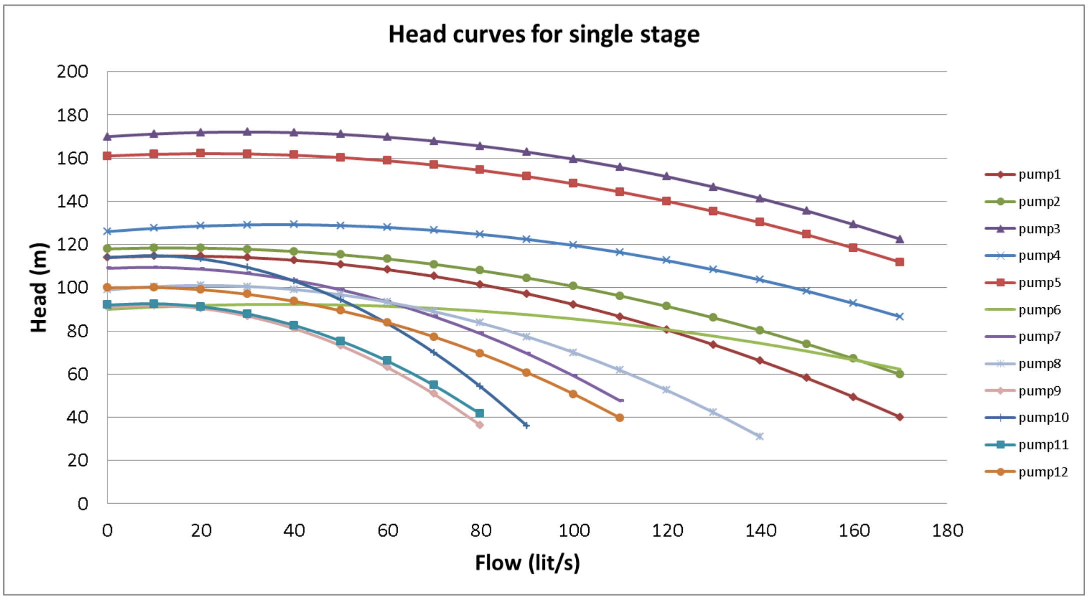


Figure 5.4 Performance curves for one stage of each pump

As the curves illustrate, a set of diverse pumps are selected to provide different combinations for optimization practices.

5.1.3. Price Data Acquisition and Estimation

The prices of the pumps presented in table 5.1 are not given in [23]. However, the prices for the pumps are estimated in this study using an internal purchasing bill for a boiler pumping system. In order to estimate the current price of the pumps, after currency unit conversion, an inflation factor is implemented, based on consumer price index (CPI). According to the statistics provided by Statistics Finland, the overall inflation factor (CPI) between 2002 (date of pricing for the reference pump) and 2013 can be determined as 1.2 times the initial prices [25]. There are different methods to estimate pump price based on an initial price. One of the well established and recognized methods for cost estimation of equipment used in process and power engineering is Walas method. Based on Walas, having base price of a centrifugal pump for a given flow and head, the change in price for higher capacities can be estimated as follows [26]:

$$C = F_M \times F_T \times C_b \times U_e \qquad [€] \qquad (5.3)$$

C = pump capital cost [€]
F_M = pump material cost factor
F_T = pump type cost factor
C_b = base cost [$]
U_e = currency unit conversion from $ to €

Considering U_e = 0.76 and F_M = 2 for stainless steel, the other two terms in Eq. 5.3 can be calculated using equations below, for a multistage pump [26].

$$F_T = \exp[9.8849 - 1.6164(\ln Q\sqrt{H}) + 0.0834(\ln Q\sqrt{H})^2] \qquad (5.4)$$

$$C_b = 1.55 \exp[8.833 - 0.6019(\ln Q\sqrt{H}) + 0.0519(\ln Q\sqrt{H})^2] \qquad (5.5)$$

In which, Q and H are capacity (gpm) and head (ft), respectively.

Equation 5.3 is useful in determination of an approximate price when there is no information beyond head and capacity at design point. However, in this study, the base price of the reference pump is known. Hence, the Eq. 5.6 seems to be more suitable for capital cost estimation [27].

$$C = C_b \times \left(\frac{P}{P_b}\right)^n \qquad [€] \qquad (5.6)$$

C = pump capital cost [€]

C_b = pump base cost [€]

P = pump power [kW]

P_b = pump base power [kW]

n = price ratio

Based on Couper, price ratio of 0.7 seems to be accurate enough for centrifugal pumps [27]. Based on the available data, having power of 2000 kW and price of 240,000 € for the reference boiler pump, the price can be estimated by change in power. After adjusting the pump prices to today's prices based on inflation rate, the prices of different pump types can be estimated for any number of the possible stages as a function of base price (Eq. 5.6). For instance, the estimated prices can be compared for 4-stage and 6-stage units (table 5.2). Pump head and power curve is also adjusted based on the number of stages.

Table 5.2 Price estimation for basic pumps and multistage pumps

	Price for 4-stage (€)	Price for 6-stage (€)
pump1	101,500	134,800
pump2	122,000	162,000
pump3	172,800	229,500
pump4	134,000	178,400
pump5	153,000	203,200
pump6	106,100	140,700
pump7	68,800	91,400
pump8	76,200	101,300
pump9	49,600	65,800
pump10	62,400	82,900
pump11	49,400	65,700
Pump12	64,600	85,700

The calculation of the VSD costs is not easy and straightforward. In absence of a precise price list and data, the cost of VSD is approximated based on the horsepower (hp) of the electric motor. It is assumed that the variable speed technique is based on

VFD and the plant has available infrastructure and place for installing VFD facilities and related systems, like air-conditioning system. Based on the data presented in [28], the average cost of VFD can be correlated to the motor size, offering 120 €/hp for middle-size and larger motors.

5.2. Modeling and Optimization

In order to define the problem and set the suitable approach for solving it, first some assumptions should be taken into consideration. This is done to ease the modeling task and to eliminate the parameters that are not highly effective compared to important variables. After defining the required variables, the general problem is decomposed to several sub-models for clarification of the effect of each parameter. This is important to monitor the effect of each variable individually, then in a group of them, to provide adequate understanding for analysis, interpretation of the results, and modifying the variables to meet the target. Then, optimization problem is defined and solved based on the available set of variables in each sub-system. However, the main assumptions presented in the previous subsections, as well as 5.2.4 are fixed in all simulation practices.

5.2.1. General Assumptions

The main objective is to model the BFPS so that it can be optimized for different conditions. First, it is assumed that the selected pumps can be used in the same system. In other words, the piping and fittings can be adjusted for suction and discharge of each pumping solution. Since the boiler applications are mainly known for high static head, the effective elements in frictional losses, e.g. pipe size, are not considered in this study. However, the general effect of frictional losses is taken into account in determination of system head curve.

It is assumed that the pumping system is used for a drum boiler with constant pressure in part load operation. Hence, the static head is assumed to be fixed in different loads.

5.2.2. Boiler Head Demand (System Head)

Based on the available load data presented in section 5.1, and considering the flow of 35 lit/s as full load capacity, the flow regime is known for entire operating range. The system head characteristics are determined based on Eq. 3.1, for the boiler plant:

$$H_{system} = H_{static} + CQ^2 = 960 + 0.0175Q^2 \quad [m] \qquad (5.7)$$

Now, having the system head curve (Eq. 5.7), full load capacity, and annual flow range the system characteristics are well known. Considering full load flow of 100 lit/s, the corresponding design head for this system is determined equal to 1135 m. In other words, if the selection was based on a single operating point, head and capacity of 1135 m and 100 lit/s were respectively the characteristics of the initial design point, for the selection of a suitable pump.

Based on the section 4.1.4 of this study, the standard requirements for this boiler plant can be summarized in table 5.3. After determination of the limits that pumps should cover, we can optimize the most cost efficient BFPS for this boiler plant.

Table 5.3 Initial design point of the system and requirements of EN standard

System requirements	Design point	EN requirements		Final set points *	
		Flow margin	head margin		
		25%	10%	1	2
Head (m)	**1135**	-	1249	**1135**	1249
Flow (lit/s)	**100**	125	-	**125**	100

* Both these points must be covered by BFPS to meet EN standard (illustrated in Fig.4.4)

5.2.3. Optimization Problem

Optimization is to find the best possible solution among a set of feasible alternatives when the favorability can be measured with some specific, numerical measure [29]. A classic optimization problem is defined as follows:

$$\min/\max_{x,y}\ f(x,y) \qquad (5.8)$$

so that:

$$g(x,y) \le C \quad \text{(inequality constraints)}$$

$$h(x,y) = D \quad \text{(equality constraints)}$$

$$a \leq x \leq b$$
$$c \leq y \leq d$$

In which, x, y are *variables* that can be changed to minimize/maximize the *objective function* $f(x, y)$. Constraint *parameters* are a,b,c,d,C, and D, which are not controllable. Theses parameters with $g(x, y)$ and $h(x, y)$, i.e. *constraint functions*, define the feasible region for the solutions. If the variables are selected from real numbers the problem is linear programming (LP), and if they are integers or binary numbers, the problem is called discrete programming (discrete optimization). More explanation will be given in solution of each optimization problem in the following sections.

5.2.4. Cost Analysis

In this study, the cost analysis and cost optimization is performed based on *energy costs* and *installation costs*. Energy cost refers to the electricity consumption of the given pumping solution, based on the price of electricity. The electricity price is 0.084 €/kWh for industrial consumer in Finland [30]. The electricity price is considered to be constant during the 10-year lifetime for all the pumps. The working hours of the pumping system is also determined from the data presented in subsection 5.1.1. Total yearly operating hours is 8400 hours for whole load range.

Installation cost includes capital cost for purchasing the pump, and VFD device in variable speed systems. In order to determine the yearly installments of the initial price for cost calculations, the following equation is used [29]:

$$YI = NPV \times r(1 - (1 + r))^T \qquad [€] \qquad (5.9)$$

YI = yearly installment [€]
NPV = net present value [€]
r = annual discount/interest rate
T = lifetime (years)

Assuming 10% interest rate, the whole capital price of the pump (and VFD if needed) is distributed in equal yearly installments. Therefore, NPV is the sum of capital investment for the pump and VFD device at present. A summary of the assumptions and data used for all scenarios are presented in table 5.4.

Table 5.4 Input data for cost calculations

	Value	Unit	Source
pump lifetime	10	years	estimation
electricity price	0.084	€/kWh	[30]
interest rate	10	%	[24]
inflation ratio (1996-2012)	1.2	-	[25]
VFD costs	120	€/hp	[28]
operating time	8400	hour/a	given
water density	0.9	kg/lit	T = 168C

5.2.5. Motor Efficiency and VFD Performance

Motor efficiency in par load operation and lower speeds is another important parameter that is often ignored in the calculation of cost savings of VSP. In other words, VSP reduces the motor speed and load in part load operation, resulting in lower electricity consumption compared to constant speed. However, the motor efficiency is also reduced in lower loads and speed. Though this reduction may not make a structural change in the results, the profit can be exaggerated if it is ignored.

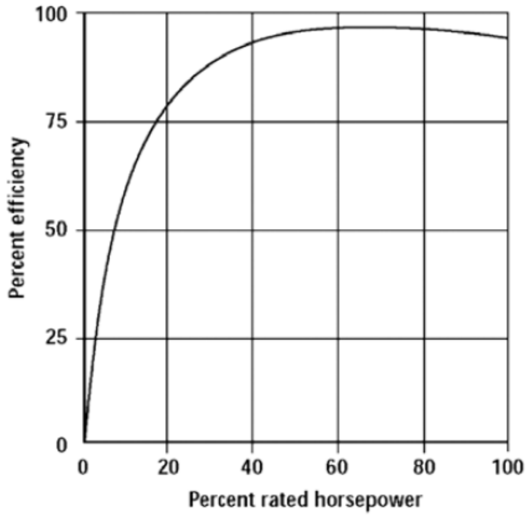


Figure 5.5 Relation of motor efficiency with motor load factor

The dependence of the efficiency of an electric motor based on load is portrayed in Fig. 5.5 [31]. This relation may be slightly different based on the size of the motor, but the general trend is similar.

The best efficiency is Moreover, the relation of the motor efficiency and changes in frequency can be explained for VFD pumping systems. Though based on [31] the precise estimation or calculation of overall motor efficiency operating under VFD is very complex, some approximations can produce a general trend. The overall efficiency of the motor can be calculated in different speeds, presented as different frequencies [32]:

$$Motor\ overall\ efficiency = VFD\ factor \times nominal\ motor\ efficiency \quad (5.10)$$

In which, nominal motor efficiency means the motor efficiency in 100% speed at specified load. VFD factor represents the inefficiencies arisen from the use of VFD device in the system. The change of idealized VFD factor (motor and VFD controller) against the frequency variations is depicted in Fig. 5.6.

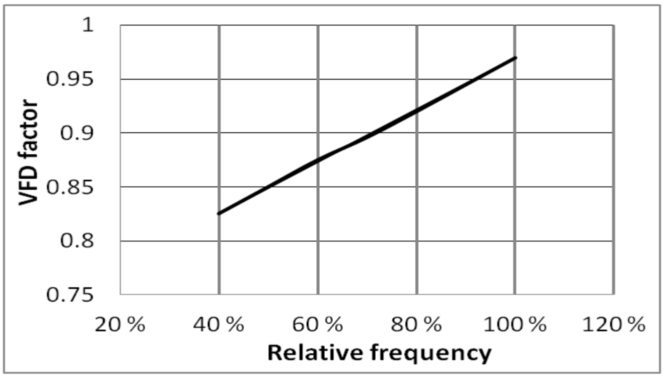


Figure 5.6 VFD factor based on change in frequency

Relative frequency shows the ratio of the frequency at variable frequency to the nominal frequency of the motor. The effects of part load operation and VFD are considered in the calculation of motor efficiency and electricity consumption of the BFPS in this study.

5.3. Scenario 1: Single Pump Optimization

As mentioned before, the problem is divided to several subsets to monitor the effect of each variable more clearly. First, the target is to select the best possible BFP (single unit) from a set of available alternatives. This approach resembles the final decision making process after receiving the proposals from different manufacturers or several options from one manufacturer. Using this approach, the difference between variable and constant speed operation will be clearly examined. Besides the pump type and pump speed, the number of stages is another variable in finding the optimized solution.

The hydraulic characteristics of the pumps are known, i.e. head, NPSH, and power curves. The initial price of each pump, price as a function of stage numbers, lifetime, electricity price, maintenance/operational costs, and interest/discount rate are also known for the calculation of the total costs. In variable speed operation, VSP costs, VFD efficiency in different loads, and speed limits are also set based on available data from the literature.

5.3.1. Problem Definition

The goal in this step is to find a single main pump that is most optimized solution for the system. A booster pump is designated to work in series with the main pump. In this step, the booster pump is fixed for all alternatives to decrease the degree of complexity of the problem. Therefore, we are just to optimize the type and stage number of the main pump in constant and variable speed in variable load. The characteristics of the booster pump which is used for this optimization problem is shown in Fig. 5.7. It is assumed that NPSH is high enough in whole flow range. According to equation 5.6, the price of this booster pump is estimated to be 102,300€. As this price is fixed for all simulation cases in constant and variable speed, it is not a variable in energy and cost calculations in this scenario.

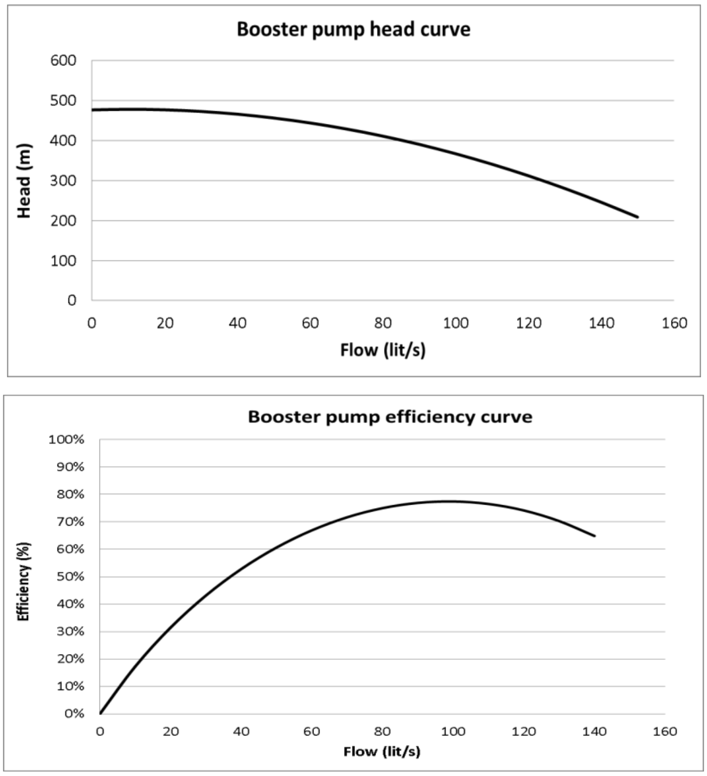


Figure 5.7 Booster pump (1) characteristics, head (above) and efficiency (below)

Among the set of available options, pumps 1 to 6 are selected for single unit operation. This is due to the wide range of flow that these pumps can supply to meet the system demand (the design flow of 100 lit/s). However, to meet the operating point and standard limits the number of stages should be optimized. As discussed before, the main goal of this scenario is to compare and analyze the role of VSP in the selection process of one single BFP.

5.3.2. Results and Discussion

After selection of six pumps (number 1 to 6 from table 5.1) for meeting the system demand, their performance and characteristics are examined and compared to find the best alternative. First, the number of stages should be determined so that the pump can handle the system demand. Here, the system demand is the system head curve given by Eq. 5.7 minus the head supplied by booster pump given in Fig. 5.6 left. The result of this head difference is shown in table 5.5 under *demand on main pump* values.

Then, having the load distribution (from Fig. 5.2), power consumption of each pump and motor size can be calculated based on the Eq. 5.2. Next, using working hours and price of electricity, the energy consumption and energy costs are calculated. Finally, using the interest rate and lifetime, the total costs are calculated for each main pump. The summary of calculations for pump 4 is presented in table 5.5. The calculations are performed for both constant and variable speed to compare the energy costs and total costs.

Based on the results, this pump is capable to meet the head demand of the system in both constant speed and variable speed. The variable speed system needs a smaller motor, resulting in lower capital cost for motor and other advantages of the use of smaller motor. The results show a higher overall efficiency in constant speed pumping (CSP) compared to VSP. This is due to the use of VFD facilities and lower load factors in lower speeds. However, the total energy consumption in VSP is much lower than CSP, 6800 and 10,300 MWh/a, respectively. It indicates that VSP offers 34% reduction in energy consumption compared to CSP for this pump.

Table 5.5 Results of the calculations for pump 5 in CSP and VSP

Pump 5 (with 7 stages)	**Constant speed**					**Variable speed**			
	Load cycle				EN load	Load cycle			
	65%	85%	Full	105%	125%	65%	85%	Full	105%
Yearly Hours (%)	30%	42%	25%	3%	-	30%	42%	25%	3%
Flow (lit/s)	65	85	100	105	125	65	85	100	105
System head (m)	1034	1086	1135	1153	1135	1034	1086	1135	1153
Booster head (m)	437	401	367	355	297	437	401	367	355
Demand on main pump (m)	**597**	**685**	**768**	**798**	**838**	**597**	**685**	**768**	**798**
Main pump supply (7-stage) (m)	1105	1071	1037	1024	965	597	685	768	798
Power (kW)	1056	1189	1287	1320	(1451)	502	716	930	1012
Energy consumption[1] (MWh/a)	2660	4193	2704	333	-	1266	2526	1952	255
Pump total energy use (MWh/a)	**9,890**				**-**	**6,000**			
Pump efficiency (%)	74 %	80 %	81 %	81 %	-	80 %	81 %	80 %	79 %
Relative speed (%)	100%	100%	100%	100%	(100%)	75 %	82 %	88 %	90 %
Motor standard power (hp)	without EN demand= 1750				2000	1750 (without EN demand, 1500)			
Motor load factor (%)	71 %	80 %	86 %	88 %	(97%)	38 %	55 %	71 %	78 %
VFD factor (%)	-	-	-	-	-	91 %	93 %	94 %	95 %
Motor efficiency (%)	97 %	97 %	96 %	96 %	(94 %)	82 %	88 %	91 %	91 %
Overall efficiency (%)	63 %	71 %	74 %	75 %	(68 %)	61 %	69 %	72 %	73 %
Total energy use[2] (MWh/a)	**10,250**				**-**	**6,840**			

1. Energy consumed by pump itself, not including motor and VFD inefficiencies
2. Including all pump, VFD, and motor inefficiency factors

Another significant result of this set of calculations is the need for larger pump and motor for meeting the EN requirements. Though this oversizing causes big losses in CSP, it may favor the use of VSP to reduce the losses. For instance, while a 1750 hp motor can properly handle the pumps for loads up to 105% of the full load, a 2000 hp motor is needed to meet the EN requirements. This oversized motor can be however useful to increase the motor efficiency. This is due to the fact that motor efficiency is higher in load factors around 75%. As the results show, having one size bigger motor gives the load factor of 88% even for pump loads higher than 100% full load.

It should be noted that the same pump assembly is used for both systems, though number of stages may be less for VSP. In other words, the difference in the output of the system is just compared for the use of VSP and CSP with exactly the same pump. Using VSP for this pump, it is possible to reduce one stage but still meeting the demands with lower costs. The total energy consumption then further reduces to 6830 MWh/a. However, lower stages (less than 6) increases the costs of VSP as the speed must exceed nominal speed to meet the demand. Therefore, the optimal number of stages for CSP is 7 while for VSP is 6 for the pump 5.

After performing the same approach for all the alternatives, the energy consumption and total costs for each pump can be calculated. The comparison among 6 BFPs is illustrated in table 5.6 in constant and variable speed. The maximum number of stages is limited to 10 stages to remain in proper boundaries for estimations.

Table 5.6 Comparing energy costs and total costs for 6 pumps in constant and variable speed

		unit	booster pump	pump1	pump2	pump3	pump4	pump5	pump6
Constant speed	stage number	-	-	10	10	6	8	7	10
	pump price	€	102300	192800	231600	229500	218100	226400	201000
	Meeting EN	-	-	No	Yes	Yes	Yes	Yes	No
	Total yearly energy costs[1]	€/a	312000	-	921200	793100	840200	861000	-
	motor size	hp	700	-	2000	2000	2000	2000	-
	Total yearly costs[2]	**€/a**	**326000**	**-**	**959000**	**831000**	**876000**	**898000**	**-**
Variable speed	Total yearly energy costs[1]	€/a	312000	550900	618300	574600	604000	574300	602700
	VFD price[3]	€	-	210000	210000	210000	210000	210000	210000
	motor size	hp	700	1750	1750	1750	1750	1750	1750
	Total yearly costs[4]	**€/a**	**326000**	**616500**	**690100**	**646100**	**673600**	**645400**	**669600**
	Notice: booster pump costs should be added to each option in calculation of the final costs of the boiler feed water pumping system								

1. Electricity costs including pump + motor efficiency
2. Including energy costs and capital costs of the pump (not motor) in 10 yearly installments
3. Based on the motor size
4. Including energy costs and capital costs of the pump + VFD (not motor) in 10 yearly installments

The first result seen in the table is the domination of energy costs in total costs for all the alternatives, with more than 89% of the annual costs in pump lifetime. It indicates that the least energy consuming option offers the least total costs. This is partly due to the high number of operating hours (8400 hours/a) for the system. Pumps 1 and 6 cannot meet the EN requirements within the number of stages so they are not among feasible solutions for CSP. However, the use of VSD facilitates these two pumps to meet the EN demand.

The most significant result from the simulation and calculations in this scenario is different optimized solutions based on speed regulation method. Among all six available options, pump 3 shows the least energy and total costs in CSP, with 793,000 and 831,000 €/a, respectively. It means pump 3 with 6-stage configuration is the optimized solution for constant speed scheme. However, among those pumps that can be used with constant speed, pump 5 has slightly better results in variable speed. Among those pumps applicable for CSP (pumps 2, 3, 4 and 5), pump 5 has the minimum energy and total costs in variable speed, 574,300 and 645,400 €/a respectively. It means if the plant owner is to select a pumping system for CSP and it is likely to upgrade the system to VSP in the future, pump 5 can be the most optimized solution.

However, pump 1 shows to be the best optimized solution for VSP among all six options, with 550,900 and 616,500 €/a for energy and total costs. In other words, while this pump cannot be used in CSP due to inability to meet the EN requirements, it is the best option in VSP. Therefore, if the BFPS is designed for running in variable speed from the beginning, the most cost efficient option is pump 1. This is the answer for one of the main questions of this study. Knowing the speed regulation method may change the optimized BFPS for a boiler plant.

Most of the pumps show the need for a smaller motor in VSP. It should be noted that the price estimation in this study may not be completely precise. The results are however reliable as the energy costs are dominant in calculation of total costs. All in all, using a VSD pumping solution for this energy-intensive application (BFPS) shows to be economical, with payback period of lower than two years. Other advantages and disadvantages of VFD should however be considered in making the final decision. Other additional costs in installation and maintenance of the pumps, motor, and VFD are not considered in this study.

The final performance curve of six pump alternatives, as well as system head curve after the booster pump is shown in Fig. 5.8. EN requirements are illustrated with cross individual points on the graph.

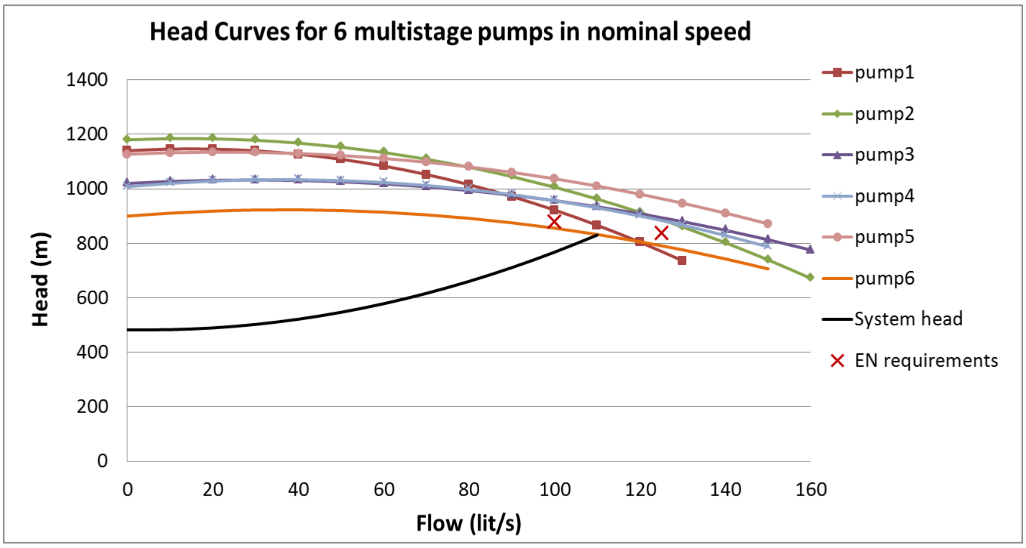


Figure 5.8 BFP head curves in CSP and system demand curve after booster pump

As we can see, pump 1 is able to handle the system demand but not for meeting EN flow requirements. One reason in difference in performance of different pumps in variable load is the change in efficiency of the pumps in different loads (Fig. 5.9).

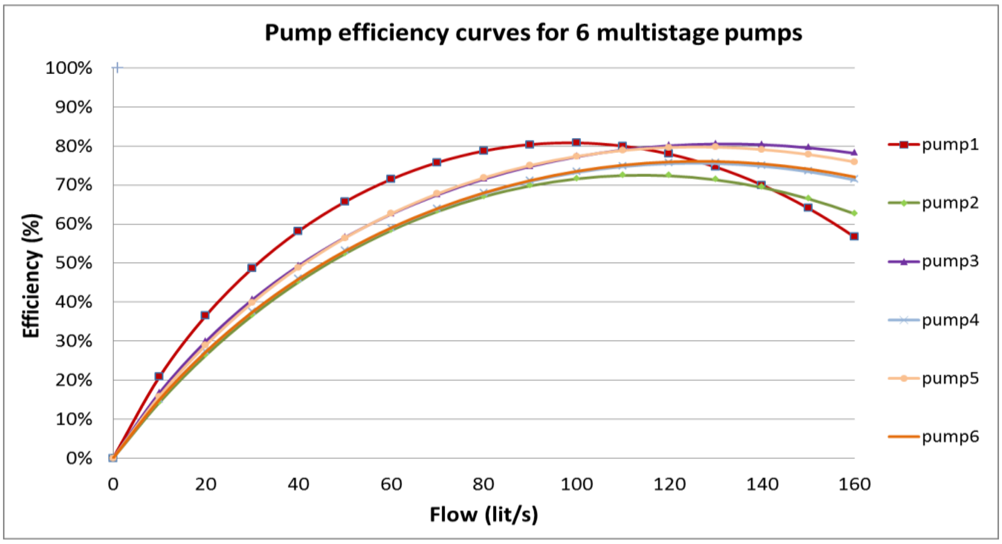


Figure 5.9 Efficiency curve for six different pumps

For example, for pump3 and 6, the efficiency increases in full load and higher loads in this load regime. However, for pumps 1, efficiency tends to increase in loads slightly less than full load, to a specific limit.

It is possible to further optimize the number of stages for each pump in variable speed. It means after selection of variable speed regulation, the pump may be smaller in size, resulting in lower energy consumption (table 5.7). Comparing the results show that it is almost possible to reduce the number of stages for variable speed.

Pump 1 shows again the minimum total operating costs among all alternatives, with 606,500 €/a. It should be noted that stage numbers lower than the optimum for variable speed will result in higher power consumption in higher loads due to relative speed values higher than 100%.

Table 5.7 Optimizing pump stage number for variable speed

		unit	booster pump	pump1	pump2	pump3	pump4	pump5	pump6
Constant speed	stage number	-	-	10	10	6	8	7	10
	pump price	€	102300	192800	231600	229500	218100	226400	201000
	Meeting EN	-	-	No	Yes	Yes	Yes	Yes	No
	Total yearly energy costs[1]	€/a	312000	-	921200	793100	840200	861000	-
	motor size	hp	700	-	2000	2000	2000	2000	-
	Total yearly costs[2]	€/a	**326000**	-	959000	**831000**	876000	898000	-
Optimal variable speed	**stage number**	**-**	**-**	**7**	**8**	**5**	**7**	**6**	**9**
	pump price	€	102300	150200	198100	202000	198600	203200	186800
	Total yearly energy costs[1]	€/a	312000	547900	617200	577400	605100	574300	605400
	VFD price[3]	€	-	210000	210000	210000	210000	210000	210000
	motor size	hp	700	1750	1750	1750	1750	1750	1750
	Total yearly costs[4]	€/a	**326000**	**606500**	683600	644500	671500	**641600**	670000
	Notice: booster pump costs should be added to each option in calculation of the final costs of the boiler feed water pumping system								

1. Electricity costs of pump + motor
2. Including energy costs and capital costs of the pump (not motor) in 10 yearly installments
3. Based on the motor size
4. Including energy costs and capital costs of the pump + VFD (not motor) in 10 yearly installments

Therefore, having the load pattern offers more sophisticated selection process. For example in this scenario (single pump optimization), if the optimization process would be performed based on a single design point, which is full-load operating point, pump 3 favors all the alternatives in CSP and VSP. However, among pumps working in both speed regulations, the results showed that pump 5 is the optimal solution for VSP in variable load (table 5.7). If pump 3 was selected without having the load data, which was the most optimized solution in that case, the extra yearly costs in VSP would be 4,000 € higher compared to pump 5. It would result in higher energy consumption equal to 370 MWh during pump lifetime.

In conclusion, using VSD offers energy savings for all four options that can be used in constant and variable speed. Pump 5 shows the highest reduction in energy consumption in variable load with 3412 MWh/a, which offers 33.3% energy savings.

For the other alternatives, using VSP has energy savings between 27% and 33%, compared to CSP.

5.4 Scenario 2: Multiple Pumps Optimization

In this scenario, two more options are introduced to the optimization process. First, the number of main pumps in parallel can be varied from 1 to 4, using identical units. Moreover, the selection of booster pump can be optimized among three available options. Hence, the final goal is to select the optimized BFPS by varying the number of stages, the number of pumps in parallel, the booster pump alternatives, and the pump type itself.

5.4.1. Problem Definition

First, three booster pumps are introduced for evaluation in different settings. The booster pumps are selected from the same source, model 3393 [23]. These booster pumps have different head and prices, to capacitate the optimization in different head demands. The performance characteristics and price estimation of these units are presented in table 5.8. Among these booster pumps, booster 1 is the same unit that was used in the first scenario (Fig. 5.7).

Table 5.8 Performance characteristics of three booster pumps

Booster pumps (model 3393) 6x8-13A RS (2 sizes) 6x8-13B RS [23]		**Head**					**Power**			
		Load cycle				EN load	Load cycle			
		65%	85%	Full	105%	125%	65%	85%	Full	105%
Flow (lit/s)		22.8	29.8	35.0	36.8	43.8	22.8	29.8	35.0	36.8
	Price (€)	Head (m)					Power (kW)			
Booster 1	85200	437	401	367	355	437	393	430	456	464
Booster 2	104000	442	426	410	405	442	489	520	544	552
Booster 3	81700	431	397	364	352	431	383	420	445	453

One of these booster pumps can be selected to contribute the main pumps to overcome the head. It is assumed that NPSHr is maintained in the whole flow range. It should be noted that these booster pumps are always in constant speed operation, even if the main pumps are in VSP mode. Among the main pumps, pumps 7 to 12 are chosen for this scenario. Therefore, the goal of optimization problem is to find the best pump type among six alternatives while optimizing the number of stages (1 to 12), the number of pumps in parallel (1 to 4), and the most suitable booster pump (one from three units).

Therefore, the optimization problem can be explained as a classic discrete optimization problem (convex). The objective function and constraints can be defined as follows:

$$\min_{n_i\,,\, m_i, s_i\,,\, m_j} \sum_{\mathrm{i}=1}^{6} \sum_{j=1}^{3} m_i n_i (CE_i + CP_i) + m_j\left(CE_j + CP_j\right) \quad [€/a] \qquad (5.11)$$

in which:

i = $\{1,2,\ldots,6\}$ main pump types (six alternatives)

j = $\{1,2,3\}$ booster pump types (three alternative)

n = number of identical pumps in parallel

m = $\{0,1\}$ binary number to control which unit is in use or not

s = number of stages of the main pumps

CE = energy costs (€/a)

CP = pump (and VFD in variable speed) capital costs (€/a)

So that (constraints):

n : positive integer number $1 \leq n \leq 4$

Only one booster pump must be selected: $\sum_{j=1}^{3} m_j = 1$

Only one main pump can be selected: $\sum_{i=1}^{6} m_i = 1$

Each pump has a minimum and maximum flow set: $Q_{min,i} < Q_i < Q_{max,i}$ [lit/s]

in which flow rate of each parallel pump (identical) is: $Q_i = {}^{Q_o}\!/_{n_i}$ [lit/s]

The sum of head of main pump and booster pump should be higher than system head in EN flow requirement(Q_{EN}):

$$\sum_{i=1}^{6} \sum_{j=1}^{3} m_i\, H_i(Q_{EN}) \; + m_j H_j(Q_{EN}) \geq \; H_{EN} \qquad (5.12)$$

in which H_{EN} is EN head requirement.

Other supporting equations that lead to the objective function can be explained as follows:

Head of each main pump: $H_i = H_i(Q_i) = s_i(\ a_i Q_i^2 + b_i Q_i + c_i)$ [m]

Head of booster pump: $H_j = H_j(Q_o) = a_j Q_o^2 + b_j Q_o + c_j$ [m]

Main pump power: $P_i = P_i(Q_i) = \ s_i\left(\acute{a}_i Q_i^2 + \acute{b}_i Q_i + \acute{c}_i\right)$ [kW]

Booster pump power: $P_j = P_j(Q_o) = \ \acute{a}_j Q_o^2 + \acute{b}_j Q_o + \acute{c}_j$ [kW]

To calculate the energy costs, the power consumption is multiplied in working hours and price of electricity. In calculation of pump capital cost for each year during the lifetime, the capital price is multiplied in installment factor, which can be obtained from Eq. 5.9. These equations are to be optimized (minimized) in objective function:

$$CE = P_i \times WH \times PE \quad [€/a]$$

$$CP = PP \times IR \quad [€/a]$$

Q_o = given duty of the whole pumping system (lit/s)

Q_i = duty of each pump working in parallel (lit/s)

a, b, c = coefficients for pump head equation

$\acute{a}, \acute{b}, \acute{c}$ = coefficients for pump power equation

WH = working hours (hr/a)

PE = electricity price (€/kWh)

PP = pump capital price (including VFD in variable speed) (€)

IR = annual installment factor (1/year)

Now, the optimization problem is properly defined. Number of possible combination of the solutions can be determined as follows:

$$\begin{aligned} &Number\ of\ possible\ solutions \\ &\qquad = number\ of\ booster\ pumps \times number\ of\ main\ pumps \\ &\qquad \times number\ of\ available\ stages \\ &\qquad \times number\ of\ possible\ parallel\ units \\ &= 3 \times 6 \times 12 \times 4 = 864 \end{aligned}$$

Among these solutions, the feasible ones should meet the constraints. Then, the optimized solution has the minimum total costs.

5.4.2. Results and Discussion

It is possible to decompose the problem to several subsets so that the solution can be reached more quickly and reliably. In this case, the optimized solution in each region is searched, and then compared to other regions to find the global solution. For example, the optimization problem can be solved for each pump individually and compared together. The summary of the results for this approach in constant and variable speed are collected in table 5.9 for pump 7.

Using pump 7, the optimized solution can be reached with 2 parallel pumps, each with 9 stages. This combination offers the minimum energy consumption and total costs for the system, when pump 7 is used. Based on the results, system overall efficiency is again higher in constant speed. Total yearly energy consumption in variable speed is 2084 MWh/a lower than constant speed, promising 28.3% savings in energy costs. The need for larger motors for meeting EN standards in any speed regulation is another result of this survey. After adaption of the same approach for each pump type, the optimization problem defined in Eq. 5.11 can be solved. The summary of the results for each pump type and the final solution are portrayed in table 5.10.

According to the results of this analysis presented in table 5.10, the optimal solution in constant speed is the same as variable speed. Pump 11 shows lower energy consumption and total costs in constant speed, 1.051 and 1.106 M€/a, respectively. In variable speed mode, pump 11 still promises to be the best choice with 0.873 M€/a for energy costs and total costs of 0.970 M€/a. Therefore, speed regulation strategy does not affect the optimal choice for boiler pumping system in this scenario. Comparing pumps’ performance in variable speed reveals that the number of pumps in parallel can be reduced for some cases, pump 9 and 10. It means while three units should work in parallel to overcome the duty in constant speed, two units are enough in variable speed. This is significant savings in capital, energy, and operating costs. The number of stages is also lower in variable speed compared to constant speed for pump 11 and 12.

Table 5.9 Results of the calculations for pump 7 in CSP and VSP

Pump 7 (2 parallel pumps, each 9 stages)	**Constant speed**					**Variable speed**			
	Load cycle				EN load	Load cycle			
	65%	85%	Full	105%	125%	65%	85%	Full	105%
Yearly Hours (%)	30%	42%	25%	3%	-	30%	42%	25%	3%
Boiler capacity (lit/s)	65	85	100	105	125	65	85	100	105
Pump capacity (lit/s)	32.5	42.5	50	52.5	62.5	32.5	42.5	50	52.5
System head (m)	1034	1086	1135	1153	1135	1034	1086	1135	1153
Optimized booster pump, head (m)	442	426	410	405	379	442	426	410	405
Demand on main pumps (m)	**591**	**661**	**725**	**748**	**756**	**591**	**661**	**725**	**748**
Main pumps supply (9-stage) (m)	954	922	892	880	827	591	661	725	748
Power (1 pump) (kW)	425	475	512	525	-	237	324	408	441
Energy consumption[1] (MWh/a)	1070	1675	1076	132	-	597	1142	857	111
Pumps' total energy use[2] (MWh/a)	2 x 3953 = 7906				-	2 x 2706 = 5412			
Pump efficiency (%)	60 %	64 %	66 %	67 %	(67%)	63 %	66 %	67 %	67 %
Relative speed (%)	100%	100%	100%	100%	(100%)	80 %	87 %	93 %	95 %
Motor standard power (1 pump) (hp)	without EN demand= 750				800	800 (without EN demand, 600)			
Motor load factor (%)	71 %	80 %	86 %	88 %	(96 %)	40 %	54 %	68 %	74 %
VFD factor (%)	-	-	-	-	-	92 %	94 %	95 %	95 %
Motor efficiency (%)	97 %	97 %	96 %	96 %	(94 %)	84 %	89 %	92 %	92 %
Overall efficiency[3] (%)	58 %	62 %	64 %	64 %	(63%)	53 %	59 %	62 %	62 %
System total energy use[4] (MWh/a)	2 x 4096 = 8192				-	2 x 3054 = 6108			

1. The sum of energy consumed by main pumps, not including motor and VFD inefficiencies
2. Based on brake horsepower for two parallel main pumps
3. Including main pumps, VFD, and motor inefficiency factors
4. For two main pumps, including inefficiency factors in main pumps, VFD, and motor

According to the results (table 5.10), the optimal solution for energy consumption may be different from the optimized solution based on total costs. For instance, pump 9 has the higher total costs while lower energy costs compared to pump 7 in VSP. Since price estimation in this study is not completely precise, the results based on energy consumption are more reliable for making final decision. Using VSP offers energy savings between 24% and 30% for different main pumps in this scenario. The highest savings in variable speed operation belongs to pump 10 with 2700 MWh/a resulting in 226,000 € cost savings per year.

Table 5.10 Optimizing pump numbers in series and parallel for constant and variable speed

		unit	pump7	pump8	pump9	pump10	pump11	pump12
Optimal constant speed	parallel pumps	-	2	2	3	3	2	2
	stage number	-	9	9	10	8	12	10
	1 pump price	M€	0.121	0.135	0.094	0.101	0.107	0.123
	booster pump	-	b2	b2	b2	b2	b2	b1
	meeting EN	-	Yes	Yes	Yes	Yes	Yes	Yes
	total yearly energy costs[1]	M€/a	1.069	1.110	1.089	1.145	1.051	1.089
	motor size[2]	hp	2400	2400	2400	2400	2400	2400
	total yearly costs[3]	M€/a	1.129	1.174	1.155	1.215	**1.106**	1.149
Optimal variable speed	parallel pumps	-	**2**	**2**	**2**	**2**	**2**	**2**
	stage number	-	**9**	**9**	**10**	**8**	**10**	**9**
	total yearly energy costs[1]	M€/a	0.894	0.934	0.890	0.919	0.873	0.905
	VFD price[4]	M€	0.192	0.192	0.192	0.216	0.192	0.192
	motor size	hp	2400	2400	2400	2600	2400	2400
	total yearly costs[5]	M€/a	0.985	1.051	0.988	1.024	**0.970**	1.012
	Notice: booster pump power and costs are included							

1. Total electricity costs of the system (booster pump + main pumps)
2. Including booster + main pumps
2. Including electricity costs and capital costs of the pumps (not motor), in 10 yearly installments
4. Based on the motor size
5. Including energy costs and capital costs of the pump + VFD (not motor) in 10 yearly installments

5.4.3 Role of Booster Pump

One of the variables in this scenario is the booster pump connected in series to the main BFPs. The booster pump can be selected from three different alternatives in optimization problem. Since these three units have different characteristics and prices (table 5.8) they can affect the optimal solution. In this study, booster pumps are always in constant speed operation even when the main BFPs are in VSP mode. The combination of parallel BFPs and booster pump is portrayed in Fig. 5.10 for pump 7. As mentioned earlier, when main BFPs cannot reach the system head curve, booster pump contribute in raising the resultant head of the pumps.

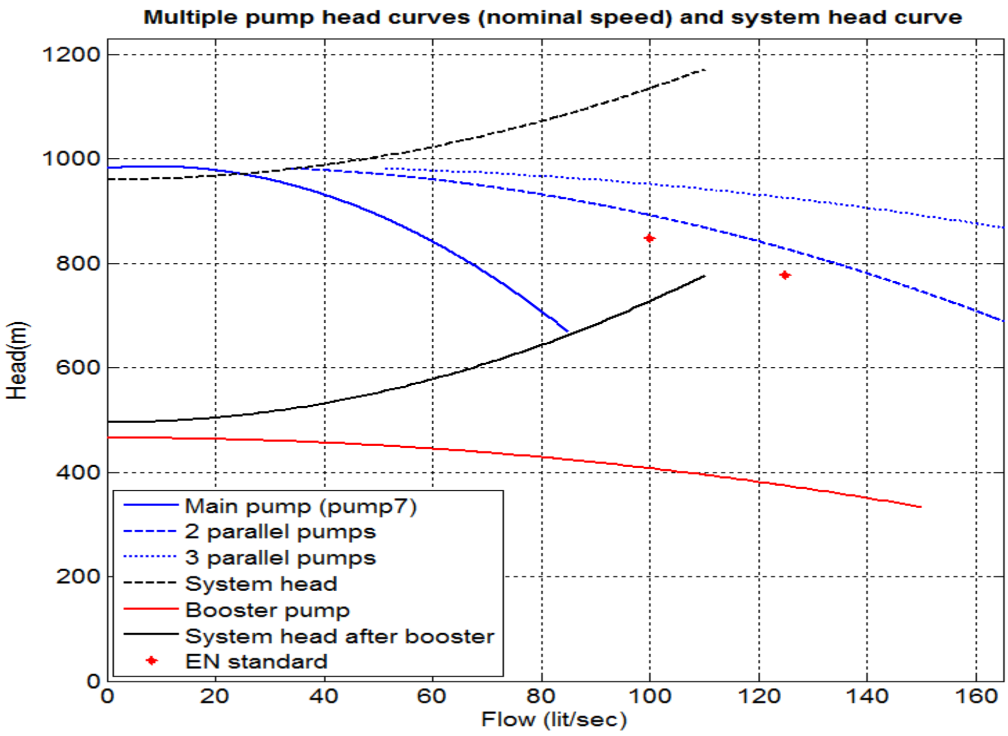


Figure 5.10 Boiler pumps in parallel, booster pump, and system requirements

5.5 Scenario 3: Optimization of Operation in Parallel

In the previous scenarios the selection process of an optimized BFPS were described. The optimal design for a single main BFP (scenario 1) and combination of BFPs in series and parallel (scenario 2) was illustrated. However, to optimize the life cycle costs of a pumping system, some measures should be considered after the initial selection. In this section, the focus is dedicated to optimal control of BFPS in operation. Here, “control” means to adapt the most suitable strategy in running the available units for different loads. There are some studies in this field discussing only two parallel units [15]. In some studies, the effect of pump’s driver is only examined [33]. There are many other studies that optimize pumping systems, based one single operational point [34], or one single pump.

In this study, two identical BFPs are studied in this scenario in series with a booster pump. It is assumed that both BFPs are driven with synchronous speed. To find the optimized control strategy, the feasible solutions should be defined based on some constraints. The constraints can be minimum flow (shut-off) and maximum flow (run-out) or some other criteria, like pump efficiency [35]. Then, based on the system characteristics, the most optimized operation method can be determined.

5.5.1. Model Construction in MATLAB

A mathematical model based on algorithm analysis is built in MATLAB for the optimization of pumps in parallel operation. Load data of the system during the operation time is imported to the code, sorted and weighted. Then, the appropriate pumps that may fir the system whether individually or in combination are introduced to the tool. The algorithm first checks the input data of each pump and synchronize them in unified units and intervals. Then, the applicability of combining pumps in parallel and series is examined. If so, head curves and power consumption is calculated in each possible combination of pumps for the entire flow range. Next, the results are produced for different speed regulation within the acceptable speed spectrum.

In each step, the ability of the BFPS to cover the system head and standard requirements is monitored. The final results are visualized in each step to provide better cognition about the optimization results and discover the drawbacks, if needed.

5.5.2. Results and Discussion

To automatically control a multi-pump system in variable speed, the speed limits should be specified. They can be based on torsional and lateral speeds provided by the manufacturer. The nominal speed of the pump and motor should also be known to define the criterion for the optimization problem. Example below deals with a BFPS including two identical pumps in parallel.

In this scenario, pump 7 is examined to establish a control strategy for parallel operation. The characteristics of one stage of this pump can be calculated using equations 5.1-2 with the data provided in table 5.1. The variable load data for the boiler is also given in sub-section 5.1.1 and Eq. 5.7. As it was mentioned in scenario 2 of this study, two units of pump 7 in parallel provide the most optimized solution in variable speed for the initial selection (table 5.10). Now, using the optimization model developed in this section, the optimized control strategy for running these two pumps can be seen in Fig. 5.11.

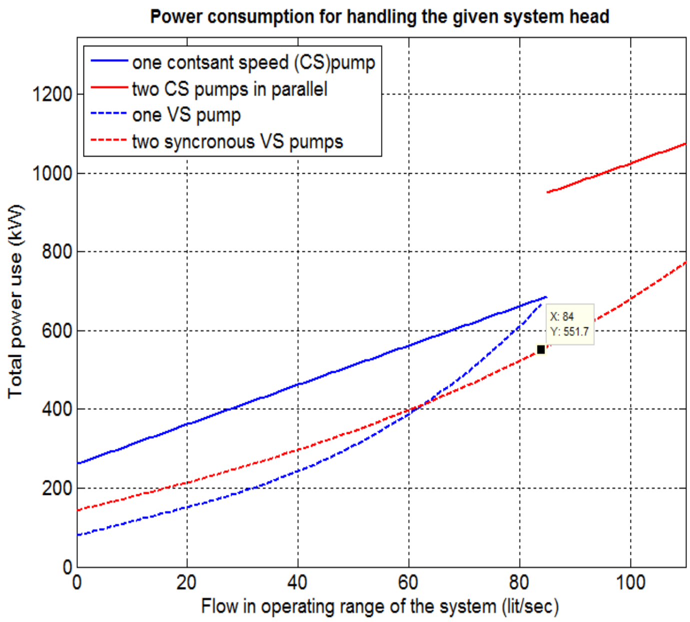


Figure 5.11 Determination of control strategy for parallel operation

According to the results illustrated in Fig. 5.11, the power consumption of one and two (parallel) pumps depends on speed regulation. The dash lines show the pump(s) brake horsepower in variable speed. This result is determined for the given pump curves and system head presented in Fig. 5.10. For the given system, one pump can be run in constant speed up to 84 lit/s, and then two pumps must be used to handle the duty. In variable speed, in single or parallel operation, the power consumption is lower but reaching the linear curves at the end, which is the nominal speed.

An important result that can be seen is the lower power consumption for parallel pumps from flow of 62 to 84 lit/s, where one single pump can also be used. It means that although one single pump can handle the task up to 84 lit/s in any speed regulation, two parallel pumps (using VSD) have lower energy consumption for the flow range between 62 and 84 lit/s. Two parallel pumps consume 550 kW at flow of 84 lit/s against 660 kW used by one single pump. It offers 465 MWh electricity cuts in one year for 50% operation, saving 38,000 € for the system. Therefore, the optimized pattern to control the operation of the pumps in parallel is the lowest intersection of all power curves, subject to minimum flow and pump efficiency limits. The right limit in figure indicates the highest flow rate in healthy operation.

6. SIMULATION IN MEDIUM-SIZE POWER PLANT

In the previous chapter, the optimal design of BFPS was studied in two scenarios: selection process and operation phase. The attention paid to find the most optimized solution from *system design* viewpoint. The result showed that combining BFPs in series and parallel can reduce the life cycle costs adapting the most suitable speed regulation for a continuously variable load. In this chapter, the performance of BFPS is further studied in a power plant to include possible influences between the pumps and other units. In other words, this chapter is dedicated to survey the change in power plant performance and output in variable load, considering the electricity consumption of the boiler pumps in different settings.

6.1. Case Study: Medium-size Power Plant

A 250MWth boiler in operation in a CHP plant is considered as the case for this analysis. The CHP plant is targeted for production of 50MW electricity and 165MW heat for the neighboring district heating (DH) network. The boiler is a drum boiler with constant pressure regulation in part load operation. The steam flow variation is assumed the same as load data presented in section 5.1. It means the effect of blowdown, desuperheating, and soot-blowing is neglected, maintaining the identical load variation pattern in BFP and turbine inlet. Hence, the operation of the plant in part load is directly translated into the load variation in BFPs. Eliminating the lateral and auxiliary units, the plant layout for the main units is portrayed in Fig. 6.1.

6.1.1. Fuel Composition

The plant is in operation in Finland. Fuel composition is taken into account to calculate the carbon emissions. The characteristics of the fuel are presented in table 6.1 for the plant in question. The missing data is acquired from [36].

Table 6.1 Fuel characteristics of CHP plant

Fuel characteristics	share	LHV_{daf}	Moisture	Dry content					
				C	H	N	O	S	ash
	%	MJ/kg	%	%dry	%dry	%dry	%dry	%dry	%dry
Wood chips	83	18.5	38	50.2	6.4	0.5	42	0.05	0.8
Peat	10	7.7	53	54.3	5.4	1.4	34.1	0.4	4.4
Coal	7	25.3	9	73.4	4.6	1.6	8.1	0.6	11.7
Mixed fuel	100	17.9	37.5	52.3	6.2	0.7	38.8	0.1	1.9

The CHP plant is equipped with a multifuel boiler benefiting from a mixture of different solid fuels, namely biomass (wood chips), coal and peat.

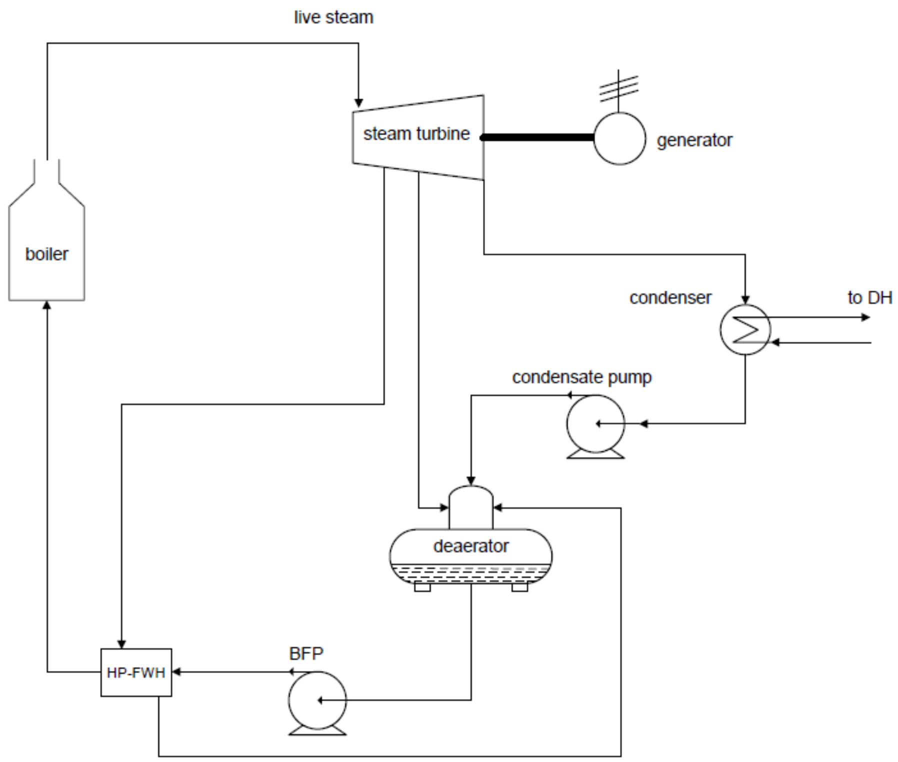


Figure 6.3 Simplified layout of CHP plant

6.1.2. Unit Parameters

The most relevant parameters of the plant units needed for simulation are presented in table 6.2. For the BFP, the duty at design operating point is known. In simulation practices, the data of the pumps examined in this study will be used when the corresponding pumping system is in question. For example, if the BFP is a single unit in constant speed, the suitable unit is selected from pump 1 to 6 presented in section 5.3 (scenario 1) while for multiple pumps from section 5.4 (scenario 2).

6.2. Modeling of Power Plant in IPSEpro

To examine the BFPS in operation in CHP plant, IPSEpro modeling tool is used in this study. IPSEpro is a equation based tool for simulation, modeling, analysis and design of components and processes in energy and process engineering [37]. IPSEpro is capable for building new models and development of the existing model library, using a Model Development Kit. In order to build the CHP plant in question, the unit parameters depicted in table 6.2 is used, as well as BFP characteristic presented in previous chapter.

Table 6.2 Equipment parameters of CHP plant

Equipment	Parameter	Unit	Value
Boiler	steam pressure	bar	92
	steam temperature	^oC	530
	steam flow	kg/s	90
	efficiency	%	92
	Air ratio	-	1.3
	pressure loss	bar	9
Turbine	inlet pressure	bar	90
	inlet temperature	^oC	525
	isentropic eff.	%	80
	mechanical eff.	%	91
Generator	electrical eff.	%	95
	mechanical eff.	%	93
Condenser	inlet pressure	bar	0.95
	DH temp.	^oC	90
Condensate pump	efficiency	%	77
	mechanical eff.	%	95
Deaerator	pressure	bar	4.85
BFP	design flow	lit/s	100
	pressure	bar	102
HP FWH	steam pressure	bar	9.5
	water pressure loss	bar	1

6.2.1. Building New Model for Pump Part Load Operation

To analysis the plant performance in part load operation, the simulations should be performed for units in part load. Among units available in model library of IPSEpro, pump and motor do not provide ability for part load modeling. Therefore, one of tasks performed in this study is to build part load models for BFP and electric motor (including VFD) so that speed regulation in variable can be studied.

For building a part load model for boiler pumps in IPSEpro, new equations should be encoded to the model library kit. These new equations are pump performance curve data, pump brake horsepower and efficiency relationship (Eq. 2.10-11), and affinity laws (Eq. 2.15-17). In the presence of geometric data of the piping, NPSH requirements can also be added (Eq. 2.15). After definition of mentioned equations, the developed head and corresponding power, efficiency, and speed regulation (in VSP) can be determined in different loads. A typical pumping system comprising of two parallel units in addition to relevant data window for newly-built pump model is illustrated in Fig. 6.2.

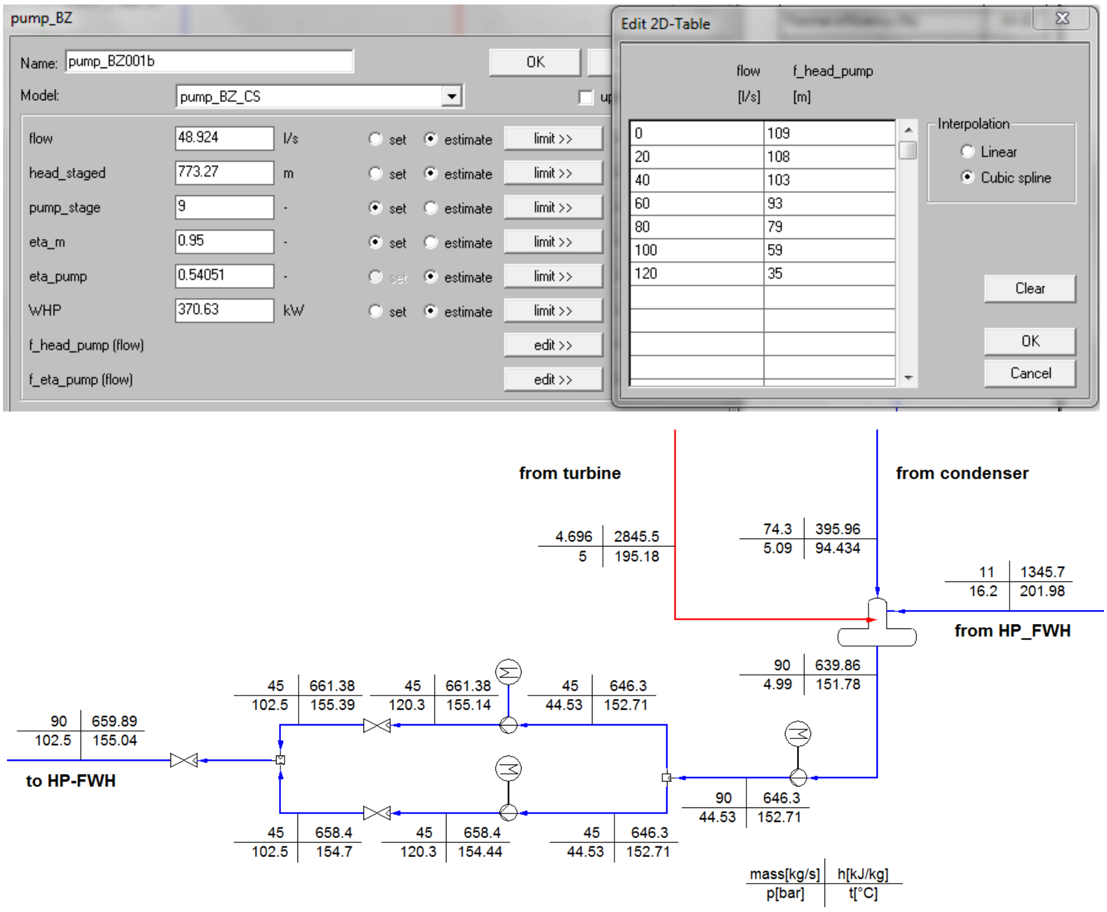


Figure 6.4 Input panel of pump model built for simulation of variable load (above) and layout of boiler feed pumping system in IPSEpro (below)

6.3. Simulation of BFP Solutions in Power Plant

After construction of new pump model in the simulation tool, the entire CHP plant can be simulated in full load and part load. New pump and motor models can incorporate the necessary characteristics of the pumping system in part load operation, as well as speed regulation. To model the CHP plant, unit parameters presented in table 6.2 is used, as well as pump data from previous chapter. Boiler model comprises economizer, steam drum, furnace, combustor (burner), circulation pump, and superheater. Using this approach, any fluctuation in parameters can be more precisely simulated. A layout of the plant in full load operation is illustrated in

Appendix 1. BFPS includes a booster pump (booster 1) and two parallel units (pump 9), which were studied earlier in second scenario.

6.3.1. Plant Output in Variable Load under Speed Regulation

Having the CHP plant model built, plant output and performance can be evaluated in different loads. Carbon emissions can be calculated for any simulation case as the composition of mixed fuel is known (table 6.1). The results of analysis of the CHP plant in part load for two pumping scheme is illustrated in table 6.3.

Table 6.3 CHP plant performance in part load for constant and variable speed boiler pumps

CHP	Load	Flow	Head demand	Pump head	Relative speed	Pumps power	Net power output	Plant electric eff.	Plant thermal eff.
unit	%	kg/s	bar	bar	%	MW	MW	%	%
Constant speed	100%	90	102	121	100	2.06	58.8	22.5	65.2
	80%	72	102	130	100	1.90	46.8	21.6	66.2
	60%	56	102	137	100	1.69	34.9	20.6	67.5
Variable speed	100%	90	102	102	90	1.60	59.3	22.7	65.1
	80%	72	102	102	83	1.36	47.4	21.9	66.0
	60%	56	102	102	78	1.13	35.5	21.0	67.2

A sample layout of the CHP plant in operation with 60% load using variable speed pumps is portrayed in Appendix 2. The required equipment parameters and assumptions are also illustrated in the same layout.

According to the results, the net electrical output of the CHP plant is higher in variable speed pumping, as expected. In 60% load, the plant offers 35.5 MWe in VSD while 34.9 MWe with constant speed BFPs. Moreover, the electric efficiency shows an increase of 2%. It offers 5370 tons reduction in carbon emissions each year. However, it is seen that thermal efficiency in constant speed is slightly higher, due to increase in feed water enthalpy before feed high pressure water heat. This additional heat is in fact the waste heat caused by throttling the extra pressure provided by pumps in constant speed.

7. CONCLUSIONS AND RECOMMENDATION

The ultimate goal of this study was to examine the optimal design of boiler feed water pumping systems, to minimize the energy consumption and total costs. By using a "system design" approach, the optimal pumping solution can be discovered by consideration of different pumps, combination of multiple pumps, and speed regulation. Boiler feed pumps are characterized for high static head that offers an approximately flat system head. Therefore, one of the main goals of the study was to examine the feasibility and economic analysis of variable speed pumping in such pumping systems.

To increase the reliability of the analysis, pump alternatives were selected from a set of boiler pumps manufactured by Goulds Pumps. The data for the water flow of the boiler pumps were obtained from an operating power plant in Finland. The flow pattern offers a variable load scheme during the year. Pump price estimation in this study is based on correlation between pump's power and capital price of a reference pump with known power. Since operating hours of the boiler pumps is relatively high, the energy costs dominate the total life cycle costs. Comparing the results in different cases shows that energy costs comprise about 90% of the yearly costs. The optimization process was divided to three different scenarios to ease the study of each variable. The detailed results are presented at the end of each section.

In utilization of one single boiler feed pump with a booster pump in series, speed regulation method and number of stages of the pumps were optimized. Variable speed in main pump showed a reduction in energy costs, up to 33%, compared to constant speed pumping. This corresponds to savings up to 285,000 € per year in energy costs, for handling the system in question. This is partly due to the variable load operation of the boiler pumps. Optimal solution can be different based on speed regulation. While some pumps showed better results in constant speed, the optimized solution in variable speed was different. Load variation pattern showed to be effective in selection of the best solution. This can be related to the fact that pumps' performance is different in different loads, based on energy efficiency curves. Therefore, knowing the load data, as well as speed regulation method before selection of the pumps can contribute in more precise optimization process.

In second scenario, the effect of multiple pumping was studied in details. The optimal solution was based on adjustment of the stage numbers, number of pumps in parallel, the most suitable booster pump, and speed regulation. The results showed that variable speed may even lessen the number of pumps in parallel from three to two. Number of stages can be 12% less in variable speed, offering lower power consumption. This further economizes the employment of variable speed facilities. However, the use of variable speed can reduce load factor, and efficiency of the electric motor in lower loads.

The power cuts caused by using variable speed in multiple pumps offered up to 28% savings in energy costs. Multiple pumps offered lower power use compared to single main pump in constant and variable speed. Using two parallel constant-speed pumps has 1,051,000 €/a as electricity costs in optimized case, which offers 5% cost reduction compared to optimal solution in case of one main pump. However, in variable speed, both single-unit and two parallel pumps have almost equal energy costs.

Not only the optimal design is applied in selection process of the boiler pumps, it should be also employed in operation control method. In this study, a mathematical model was developed to examine the most optimized strategy in controlling the pumps in parallel operation to minimize the costs. The results illustrated that energy consumption can be offset by controlling pumps far enough from low efficiency operation. For instance, using two identical pumps in synchronous speed can consume less electricity rather than using only each of them for handling the flow, in some specific ranges. This specific flow range can be discovered by using the algorithm developed in this study.

Then, a multiple boiler pumping system was analyzed in a multi-fuel CHP plant operating in different loads. The results agreed with the previous findings in energy savings caused by variable speed. However, the difference in power plant efficiency is not proportionally significant. This is due the heat passed to the water in throttling method in constant speed. Variable speed pumping can lower the carbon emissions up to 5400 tons during the year, in a 300MWth CHP plant with biomass, peat, and coal as fuel.

In conclusion, this study illustrates that the use of variable speed pumping can be beneficial in boiler applications. This depends on the degree of variation in load, and operating hours in each load segment. This is also due to the fact that meeting EN standards leads the boiler pumping systems to be oversized. Therefore, the pump is in permanent part load operation, even maintaining the full load demand of the boiler. However, other benefits and drawbacks of variable speed pumping should be carefully considered in making the final decision for a given boiler pumping system.

The future study can be focused on optimal control method in operation of two or more boiler pumps in parallel in asynchronous speed. Development of variable speed control equipment has provided new technological opportunities in running parallel pumps in different speeds. The application of variable to booster pump can be another topic for further examination. From system design viewpoint, the proportion by which the load can be divided between multiple pumps, in series and parallel, based on speed regulation method is an important step to improve system energy efficiency. This approach is applied for boiler pumps in this study, but can be further used for other pumping systems. The optimized speed regulation algorithm for pumps in operation can be also applied to other systems with more complicated characteristics.

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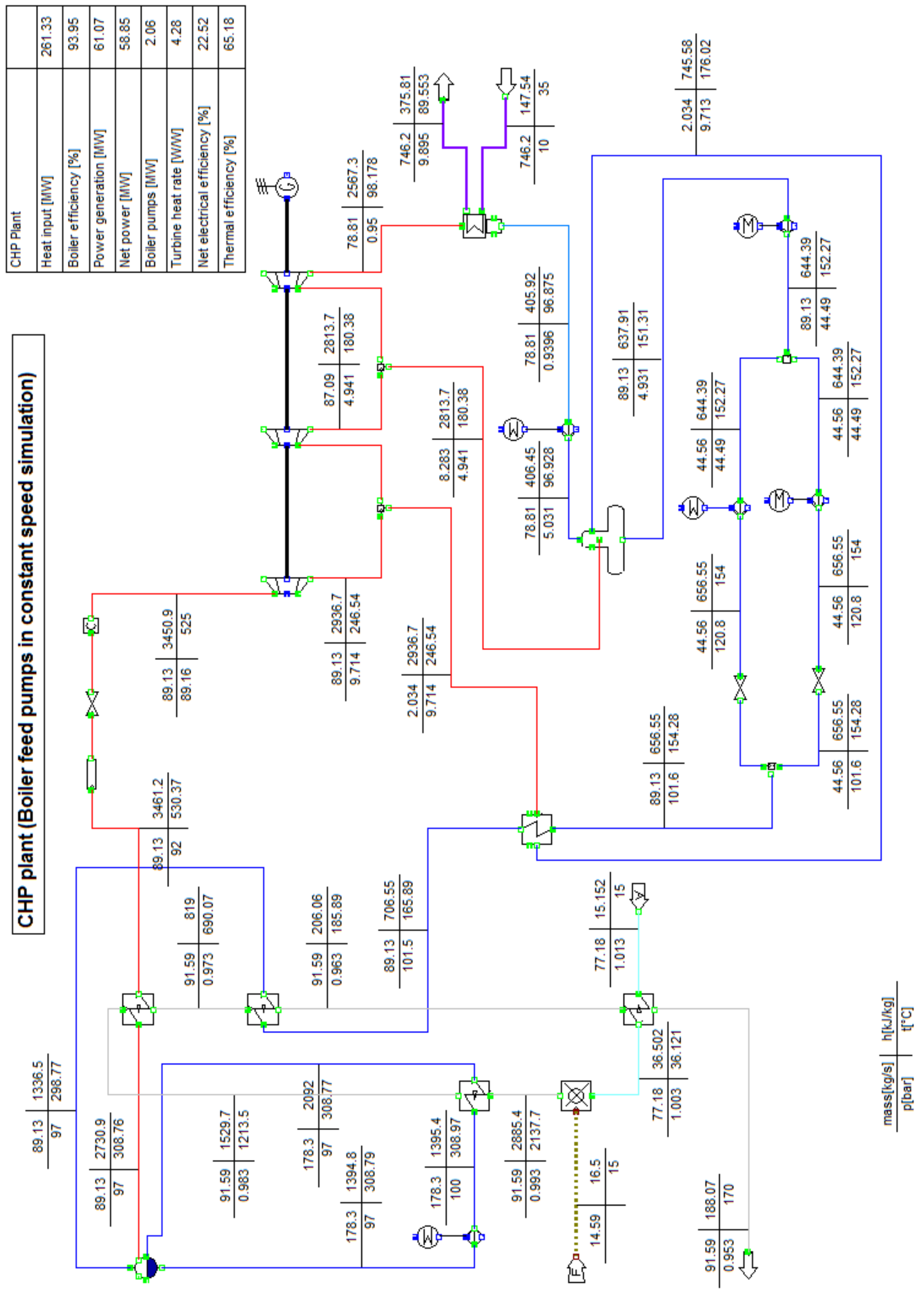
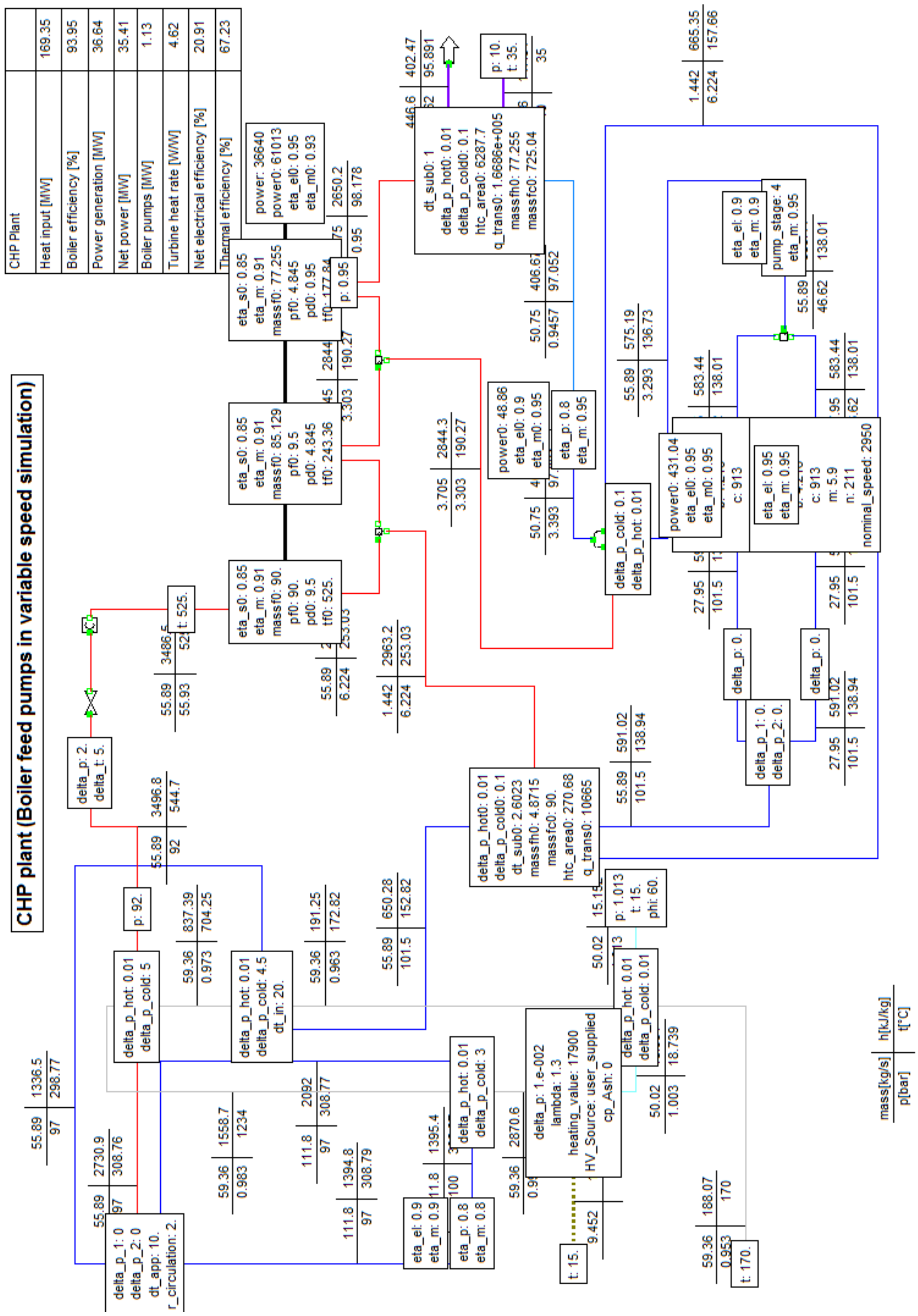
APPENDIX

Appendix 1: CHP plant layout in full load operation using two parallel boiler feed pumps and a booster pump



Appendix 2: CHP plant layout in 60% load operation using two parallel boiler feed pumps in variable speed and a booster pump in constant speed



LIITE IV

Varapumpun tarpeellisuus – projektiehdotus 20.3.2014

VARAPUMPUN TARPEELLISUUS

Soodakattilayhdistys on jo pitkään ihmetellyt monia soodakattilan erityispiirteiden teknisiä vaatimuksia kuten miksi vaaditaan varapumppu. Tähän samaan kokonaisuuteen kuuluu säännöstö millä määritellään kattilan hätätyhjennys.

Esitetään että aloitettaisiin yhdistyksen omana projektina tutkimaan asioita. Mukaan haluttaisiin TUKES ja vakuutusyhtiöt.

Tavoite olisi saada aikaan selvennetyt ja johdonmukaiset suunnitteluohjeet

- syöttövesipumppujen valinnalle erityisesti varapumpulle (turbopumppu)
- veden pinnalle kun kattila hätäpysäytetään
- ym.

Taustaa:

Soodakattila on suunniteltu siten että se voidaan tarvittaessa tyhjentää ja ajo lopettaa. Tällöin veden pinta lasketaan pari metriä pohjan alinta tasoa korkeammalle. Tämä on poikkeuksellista kattiloissa. Soodakattila ei myöskään vaurioidu sitä tyhjennettäessä. Varapumpun tarvetta taas perustellaan sillä että veden pintaa pitää pitää lieriössä näkyvillä, koska muutoin kattila vaurioituu. Tässä on ristiriita.

Veden pinnan korkeutta kun kattila hätäpysäytetään on pohdittu, mutta löytyykö asiasta hyvä tietoa?

TUKESin kanssa voitaisiin avoimesti keskustella ja pohtia mitä toimenpiteitä ja tutkimuksia tarvitaan jotta selvyys asiaan saadaan.

LIITE V

ÅA, Black Liquor Evaporation Book
– projektitarjous 20.3.2014

Proposal: Black Liquor Evaporation Book

Prepared by: Nikolai DeMartini

This is a proposal to write a book on black liquor evaporation with Jim Frederick. This proposal to SKY is for covering Niko DeMartini's portion of the work. The money for Jim Frederick's time is being proposed to AF&PA. The Table of Contents for the proposed book is attached. The book provides introductory text to the basic principles of black liquor evaporation for young engineers in chemical recovery and detailed black liquor properties and scaling chapters which will serve as both a reference, and as a guide to troubleshooting and resolving scaling problems. This keeps the scope within the areas of expertise of the authors and also allows us to deliver the completed manuscript by 30 June 2015.

Black liquor evaporation will likely be affected by lignin removal in a number of ways. Boiling point, viscosity, solubility limit and the first salt to precipitate may all be affected by lignin removal. This book will include the available public knowledge on the properties of reduced lignin black liquor and evaporator scaling in evaporation of reduced lignin black liquor. Additionally, the available scaling and fouling knowledge can be utilized to predict potential changes in the scaling behavior of reduced lignin Kraft black liquor. A first estimate of solubility limits can be made from the literature and the salt expected to precipitate can be determined from the CO_3/SO_4 ratio. These can be compared to results from Chalmers which will be presented at the ICRC and included in the literature review. This information will provide a first indication of how black liquor evaporation load should be adjusted, in particular to avoid upsets due to fouling. This is a small, but important part of the preparations for full scale implementation of lignin removal.

The cost for Niko's part in this proposed work is 14 750€ not including VAT. This includes a 1000€ honorarium for a European reviewer as well as some money for travel/reporting in Finland. We would suggest Lars Olauson. The proposal to AF&PA will include money for a reviewer in N. America. We would suggest David Clay. This proposal does not include publishing. We will work with SKY and AF&PA to find an agreeable method of publishing. The timeline can be adjusted if necessary to accommodate spreading the budget over two years for example.

PROCESS CHEMISTRY CENTRE

ÅBO AKADEMI UNIVERSITY

Table of Contents

5 Research needs in black liquor evaporation

LIITE VI

ÅÅ, Single Droplet Combustion and Single Particle Modeling of Reduced Lignin Black Liquor – projektitarjous 20.3.2014

Proposal: Single Droplet Combustion and Single Particle Modeling of Reduced Lignin Black Liquor

Prepared by: Nikolai DeMartini; Anders Brink; Mikko Hupa

Single particle experiments allow us to measure a number of different combustion characteristics which can be compared with other liquors and in that way provide some basic knowledge about how reduced lignin black liquor is expected to burn in a recovery boiler: swelling, combustion times, NO and cyanate formation and sulfur release. This data can also be used in a single particle model and in CFD models. This is particularly important in understanding NO emissions, carryover, char burn-out, and sulfate reduction.

Niklas Vähä-Savo will be presenting results for a softwood Kraft black liquor and a eucalyptus Kraft black liquor as well as a soda liquor at the ICRC. This work would include additional work with the softwood liquor around sulfur and nitrogen release during pyrolysis. It also includes mixing the hardwood and softwood liquors to study the resulting mixed liquor because they behaved differently when it came to swelling. This work will also include money for additional analysis to complement the analysis we have for the softwood and hardwood black liquors.

In the modeling portion of this work we will improve our single particle model to understand the influence of temperature, oxygen concentration and droplet size on droplet conversion; sulfur release and sulfate reduction; nitrogen release and cyanate formation. This will involve adding sulfur and nitrogen chemistry to the single particle model and utilize the single particle results.

The cost for this work as proposed is **15 000**€ not including VAT. The scope/cost can be adjusted as needed as can be the choice of black liquor.

LIITE VII

ÅA, CFD Modeling of Reduced Lignin Black Liquor Combustion – projektitarjous 20.3.2014

Proposal: CFD Modeling of Reduced Lignin Black Liquor Combustion

Prepared by: Nikolai DeMartini; Markus Engblom; Anders Brink Mikko Hupa

Lignin extraction from black liquor will result in some operational changes in pulp mills. There is a lack of publically available experience in firing black liquor in a recovery boiler. This work will support full scale implementation of the technology. We will prepare a follow up proposal for a measurement campaign in a mill implementing lignin removal if SKY decides to work together on mill implementation of this technology.

For combustion, important questions remain about the spraying of the black liquor into the recovery boiler and the heat and carbon distribution in the lower furnace. These variables are important to sulfate reduction, char burnout, sulfur release and the nitrogen chemistry. In this work CFD modeling will be used to provide an understanding of how key variables such as black liquor spraying temperature and air distribution will affect the black liquor char distribution, carryover and how air distribution might need to be changed.

We will use and existing CFD model for an existing Finnish recovery boiler. We will make calculations for one black liquor at two levels of lignin reduction. The levels of lignin reduction suggested are 10% and 20%, but will be agreed upon with SKY. The first two runs will be with two levels of lignin reduction using the same firing conditions as for the original black liquor. The total solids fired will be reduced based on the level of lignin reduction and the air flow will be adjusted accordingly to maintain the same level of excess O_2. The fraction of air added at the primary, secondary and tertiary air levels will remain the same for these first runs. This will provide a carbon distribution in the lower furnace and a temperature distribution in the furnace. Two additional cases will be run for each level of lignin reduction. The first variable to adjust will be the air distribution according to the carbon distribution between devolatilization and char burning. The second variable will be a change in liquor firing to place more black liquor solids on the walls and char bed to match the distribution of the base case. These cases will provide a framework from which changes to firing patterns can be determined.

This work would be completed by spring of 2015 and is considered a pre-study that could support full scale implementation of the technology. It will provide information about how lignin removal and operational variables will affect recovery boiler performance. This information can be utilized to help make plans for how to adjust recovery boiler operation to account for lignin removal for a smoother integration of the new technology. A proposal will be prepared in a timely manner for measurements in the recovery boiler and black liquor evaporators at a mill implementing lignin removal if SKY agrees to work together for looking at the impact of lignin recovery on mill operations.

The cost for this work as proposed is **15 000**€ not including VAT. The scope/cost can be adjusted if needed. For example, if a mill is identified, it would be possible to create a baseline recovery boiler CFD model based on that mill’s recovery boiler, but this would take more time and be more expensive than the use of a currently existing base case. It would, however, have the advantage of being directly applicable to a mill campaign.

LIITE VIII

Muiden työryhmien kuulumiset

Muut työryhmät

Opinnäytetyöpalkinto

- Opinnäytetyön tulee olla valmistunut 1. kesäkuuta 2013 ja 31. toukokuuta 2014 välisenä aikana.
- Sulfaattisellutehtaan talteenotonalueelta
- Yksi hakemus tähän mennessä:
  - Fanni Mylläri, TTY: Gas–particle equilibrium of alkali metal compounds studied in an aerosol test reactor
- Levittäkää tietoa omissa yrityksissä

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

SKY 50-vuotta ja ICRC 2014

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

SKY 50-vuotta ja ICRC 2014

- Yhdistys järjestää 50-vuotisjuhlien yhteydessä ICRC-seminaarin kesäkuussa 2014 (9.-13.6.2014) Tampereella.
- Juhlatoimikunta vastaa keskiviikon 11.6 ohjelmasta
- Muiden päivien (ma,ti, to) ohjelmasta vastaa ICRC:n työryhmät
- Konferenssin nettisivut http://www.soodakattilayhdistys.fi/secure/ICRC/ICRC_index.html

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

SKY Historiikki

- Tekijä Julkaisutuotanto Risto Valkeapää:
  - kirjoittanut useita historiikkeja mm. ETY, Porvoon osuuspankki ja Porvoon Energia, julkaisee lisäksi Enertec-lehteä
  - jäämässä eläkkeelle maaliskuussa
- Alustava sisältösuunnitelma:
  - Työryhmien esittely; keitä niissä on nyt ja on ollut, mitä ne tekevät ja mitä ovat saaneet vuosien saatossa aikaan. Aineisto tulisi puheenjohtajahaastatteluista ja pöytäkirjoista.
  - Kun työryhmät on käyty läpi, tarkempi kirjan sisältösuunnitelma on saanut substanssipohjaa.
  - Toinen työryhmäesittelyn rinnalla kulkeva strategia on haastatella viirimiehet vanhimmasta päästä alkaen

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

Konemestaripäivät 2014

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

Konemestaripäivät 2014

- Järjestettiin 12-13.2.2013 Scandic Vierumäellä
- Tehdasvierailu SE Heinola
- Ohjelma (5 kpl, 30 min)
  - Suomen vauriot ja hajukaasusuositus: Markus Nieminen
  - Ruotsin vauriot: Sven Lahti, SBL Engineering
  - Maailman suurimman soodakattilan pohjan tyhjennys suolasulasta: Timo Karjunen, Varo Oy
  - Valmet Precipitators for P&P and PG: Juha Tolvanen, Valmet Power Oy
  - Korroosio soodakattilan lämpöpinnoilla, Keijo Salmenoja, Andritz Oy

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

KTR: Vauriot

- 1/2013, MF Joutseno, sulavuoto primääri-ilmakanavistoon
- 2/2013 MF Äänekoski, tulistinkorroosio
- 3/2013 MF Äänekoski, konvektio-osa,
- 4/2013 SE Oulu, ekonomaiseri
- 5/2013 UPM Kaukas, ekonomaiseri
- 6/2013 UPM Kaukas, ekonomaiseri
- 7/2013 UPM Kaukas, ekonomaiseri
- 8/2013 Kotka Mills, tulipesä/takaseinä
- 9/2013 SE Sunila, tulistin
- 10/2013 SE Sunila, tulipesä, pohja

KTR: Aktiivihiilen mitoituksen varmistus ja optimointi sekä TOC-reduktion varmistaminen, JPAnalysis / Oulun Yliopisto

- Tarkoitus varmistaa suodattimen mitoitus koeajoilla eri virtaamilla ja selvitetään aktiivihiilisuodattimen kustannukset ja toimittajat. Lisäksi vertaillaan kahden TOC-laitteiston antamia tuloksia
- Raportti saatu -> Reijo Hukkanen kommentoi ennen julkaisua
- Oulun tehtaalla suunnitellaan teollisuusmittakaavan aktiivihiilisuodatinta ioninpoistosarjaan anionivaihtimen jälkeen ennen sekavaihdinta. Vesi-Pauli Oy:n hinta-arvio on noin 200 000 euroa. Hiilen määrä noin 10 m^3.

KTR: TOC-mittaukset MF Kemi

- Tavoite:
  - Selvittää MF Kemin tehtaan soodakattilan ioninvaihtosarjan yhteydessä olevan UV-laitteiston TOC-reduktiotehoa
- Tulokset
  - UV-laitteen päällä ollessa sekavaihtimen jälkeen TOC-pitoisuus oli noin 30% (154 ppm -> 110 ppm) alempi kuin ilman UV:ta mikä vastaa suunnilleen aikaisempia tutkimuksia.
  - Näytteidenotto hetkellä UV-laitteiston kaikki lamput oli toiminnassa ja juuri vaihdettu.
  - Lisäksi orgaanisen hiilen määrä lisääntyy höyrykierrossa eli konsentroituu, mittausten keskiarvo 405 ppm

KTR: Suojaussuosituksen päivitys

- Kappale 1 Soodakattila materiaalit ja hitsaukset (KTR):
  - Sihteeri päivittänyt vanhan materiaalin ja teksti käyty läpi KTR:n kokouksessa 14.11.2013, puuttuu teksti päällehitsauksesta
- Kappale 2 Soodakattilapinnoitukset (VTT)
  - VTT päivittänyt pinnoiteosuuden. Osuus hyväksytty 23.1.2013
- Kappale 3 Paineastian korjaukset (KTR)
  - Sihteeri on päivittänyt kappaleen 3 vastaamaan tämän hetken tilannetta. Käytiin läpi ja hyväksyttiin kokouksessa 8.9.2011
- Kappale 4 Soodakattilatarkastukset (Inspecta)
  - Inspecta on päivittänyt suosituksen tarkastusosuuden. Käytiin läpi ja hyväksyttiin kokouksessa 8.9.2011.
- Kappale 5. Soodakattilan vauriot (KTR)
  - Sihteeri kerää viimeisen kymmenen vuoden ajalta tyypillisiä vauriota tietokannasta. Kommentoidaan KTR:n kokouksissa.

YTR: Sähkösuodintuhkan hyötykäyttö, Sirra

- Sellutehtaan ylijäämäisen rikkitaseen hallitsemiseksi on tapana liuottaa lentotuhkaa ja viemäröidä se jäteveden mukana. Ympäristölupien uusinnan yhteydessä tehtaille voi tulla rajoituksia tähän käytäntöön
- Tuhkan vieminen kaatopaikalle ei ole sallittua koska tuhka ylittää kaatopaikkajätteelle määritellyn liukoisuusrajan, joten keinot päästä eroon tuhkasta on liuotus tai hyötykäyttö.
- Projektin tavoite:
  - Aikaisemman suodintuhkan puhdistushankkeen tuloksien päivittäminen nykyiseen markkina- ja hintatilanteeseen sekä sähkökemiallisen käsittelymenetelmän käyttökelpoisuuden arviointi

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

YTR, BAT-dokumentin kommentointi

- Muutoksia verrattuna alkuperäiseen ehdotukseen:
  - Päiväkeskiarvot poistettu, paitsi SO2/TRS
  - CO-raja poistettu
  - NOx-raja saatu nostettua
- Aikataulu:
  - TWG (technical working group) kokous pidetty huhtikuussa 2013
  - PreFinal draft julkaistu toukokuussa 2013
  - BAT Foorumin kokous, syksy 2013
  - Sitten EU komission hyväksyttäväksi
  - Julkaisu Q2/2014?
  - Tehtaiden lupaehdoissa viimeistään 4 vuoden sisällä eli Q2/2018

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

YTR, POPE

- Soodakattilan mittaukset tehty Kymillä loppuvuonna 2012
  - Mittaukset sekä pesurin että sähkösuodattimen jälkeen
  - Savukaasupesuri näyttää poistavan hiukkasia suhteellisen tehokkaasti
  - Alustavien toksisuustulosten mukaan pesurin jälkeen korkeammat kuin sähkösuodattimen jälkeen
  - Pesuri pesee pois suolaa, konsetraatio isompi?
  - Pesuvesi Kymijoesta, epäpuhtauksien vaikutus? Vesinäytteen tutkimus ei onnistu pesurista -> rutiinimenetelmillä ei onnistu Kymen laboratoriossa.
  - Pesuri poistaa NO2 (5-10% kokonais NOx)
  - Soodakattilan PAH-päästö viisinkertainen pellettikattilaan verrattuna. Pelkistävät olosuhteet?
  - Meesauunimittausten tulokset puuttuu
- Jatkotutkimus?

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

Hajukaasusuosituksen päivitys

- Päivitystyö valmis, suomenkielinen suositus julkaistu 30.10.2014
- Englanninkielisen käännöksen läpikäynti tekemättä
- Suosituksen laajentaminen keräilyyn?
  – Räjähdykset tapahtuneet keräilyn puolella, mm. hakesiilo ongelmallinen
  – Millä työryhmällä?

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

ATR: Ohje UPS-järjestelmän periaatteeksi

- Pöyry tehnyt ATR:n kanssa ohjeen
- Ohjeen sisällysluettelo:
  – 1. Johdanto
  – 2. Sähkökatkoksen vaikutukset soodakattilan toimintaan
  – 3. Varmennetuista verkoista syötettävät kuormat
  – 4. Suositeltavat UPS verkon rakenteet
  – 5. UPS verkon suunnittelussa huomioon otettavia näkökohtia
  – 6. UPS laitevalinnassa huomioitavia seikkoja
  – 7. Suosituksia testauskäytännöistä ja kunnossapidosta
  – 8. Muut standardit ja suositukset

SUOMEN SOODAKATTILAYHDISTYS
FINNISH RECOVERY BOILER COMMITTEE

Projektiehdotuksia

- Sularännit, käyttöongelmat ja soodasulan juoksevuus
- Kustannustehokkain tapa poistaa ammoniakkia talteenottokierrosta
- IE-direktiivin vaikutukset soodakattilan päästörajoihin
- Suolallisen mustalipeän kuivaaineen määritys ja tuloksen ilmaisu
- Lipeäkierron kemiallisten parametrien optimointi
- Lipeäkierron typpi
- Ääninuohouksen mahdollisuudet soodakattilassa

LIITE IX

Euroopan parlamentin ehdotus tiettyjen keskisuurista polttolaitoksista ilmaan joutuvien epäpuhtauspäästöjen rajoittamisesta

EUROOPAN UNIONIN NEUVOSTO

Bryssel, 23. joulukuuta 2013 (OR. en)

18170/13 ADD 1

ENV 1236
ENER 601
IND 389
TRANS 694
ENT 357
SAN 557
PARLNAT 326
CODEC 3089

SAATE

Lähettäjä:	Euroopan komission pääsihteerin puolesta Jordi AYET PUIGARNAU, johtaja
Saapunut:	20. joulukuuta 2013
Vastaanottaja:	Uwe CORSEPIUS, Euroopan unionin neuvoston pääsihteeri
Kom:n asiak. nro:	COM(2013) 919 final Annexes 1 to 4
Asia:	LIITTEET ehdotukseen Euroopan parlamentin ja neuvoston direktiiviksi tiettyjen keskisuurista polttolaitoksista ilmaan joutuvien epäpuhtauspäästöjen rajoittamisesta

Valtuuskunnille toimitetaan oheisena asiakirja – COM(2013) 919 final Annexes 1 to 4

Liite: COM(2013) 919 final Annexes 1 to 4

EUROOPAN
KOMISSIO

Bryssel 18.12.2013
COM(2013) 919 final

ANNEXES 1 to 4

LIITTEET

ehdotukseen

EUROOPAN PARLAMENTIN JA NEUVOSTON DIREKTIIVI

tiettyjen keskisuurista polttolaitoksista ilmaan joutuvien epäpuhtauspäästöjen rajoittamisesta

LIITTEET

ehdotukseen

EUROOPAN PARLAMENTIN JA NEUVOSTON DIREKTIIVI

tiettyjen keskisuurista polttolaitoksista ilmaan joutuvien epäpuhtauspäästöjen rajoittamisesta

LIITE I

Tiedot, jotka toiminnanharjoittajan on ilmoitettava toimivaltaiselle viranomaiselle

1. Keskisuuren polttolaitoksen nimellinen lämpöteho (MW);

2. Keskisuuren polttolaitoksen tyyppi;

3. Käytettyjen polttoaineiden tyyppi ja osuus liitteessä II vahvistettujen polttoaineluokkien mukaisesti;

4. Keskisuuren polttolaitoksen käyttöönottopäivä;

5. Toimiala, jolla keskisuurta polttolaitosta käytetään, tai laitoskokonaisuus, jossa sitä käytetään (NACE-koodi);

6. Keskisuuren polttolaitoksen odotettu käyttötuntien määrä ja keskimääräinen kuormitus käytössä;

7. Sovellettavat päästöjen raja-arvot ja toiminnanharjoittajan allekirjoittama vakuutus siitä, että laitosta käytetään kyseisten raja-arvojen mukaisesti asianomaisesta 5 artiklassa tarkoitetusta päivästä;

8. Jos 5 artiklan 2 kohdan toista alakohtaa sovelletaan, toiminnanharjoittajan on annettava allekirjoitettu vakuutus siitä, ettei laitosta käytetä yli 300 tuntia vuodessa;

9. Toiminnanharjoittajan nimi ja sääntömääräinen kotipaikka sekä kiinteiden keskisuurten polttolaitosten osalta osoite, jossa laitos sijaitsee.

LIITE II

Direktiivin 5 artiklan 1 kohdassa tarkoitetut päästöjen raja-arvot

Kaikki tässä liitteessä vahvistetut päästöjen raja-arvot on määritettävä 273,15 K:n lämpötilassa ja 101,3 kPa:n paineessa ja savukaasujen vesihöyrypitoisuuden mukaan tehtävän korjauksen jälkeen sekä standardoituna happipitoisuuteen, joka on kiinteitä polttoaineita käyttävien polttolaitosten osalta 6 prosenttia, nestemäisiä ja kaasumaisia polttoaineita käyttävien muiden polttolaitosten kuin moottoreiden ja kaasuturbiinien osalta 3 prosenttia sekä moottoreiden ja kaasuturbiinien osalta 15 prosenttia.

1 osa

Olemassa olevien keskisuurten polttolaitosten päästöjen raja-arvot

1. Muiden keskisuurten polttolaitosten kuin moottoreiden ja kaasuturbiinien päästöjen raja-arvot (mg/Nm^3)

Epäpuhtaus	Kiinteä biomassa	Muut kiinteät polttoaineet	Muut nestemäiset polttoaineet kuin raskas polttoöljy	Raskas polttoöljy	Maakaasu	Muut kaasumaiset polttoaineet kuin maakaasu
SO_2	200	400	170	350	-	35
NO_X	650	650	200	650	200	250
Hiukkaset	30([1])	30	30	30	-	-

([1]) 45 mg/Nm^3 niiden laitosten osalta, joiden lämpöteho on enintään 5 MW.

2. Moottoreiden ja kaasuturbiinien päästöjen raja-arvot (mg/Nm^3)

Epäpuhtaus	Laitostyyppi	Nestemäiset polttoaineet	Maakaasu	Muut kaasumaiset polttoaineet kuin maakaasu
SO_2	Moottorit ja kaasuturbiinit	60	-	15
NO_X	Moottorit	190 ([1])	190 ([2])	190 ([2])
	Kaasuturbiinit ([3])	200	150	200
Hiukkaset	Moottorit ja kaasuturbiinit	10	-	-

([1]) 1850 mg/Nm^3 seuraavissa tapauksissa:

(i) diesel-moottorit, joiden rakentaminen aloitettiin ennen 18.5.2006;

(ii) kaksoispolttoainemoottorit nestemäisen polttoaineen moodissa.

([2]) 380 mg/Nm^3 kaksoispolttoainemoottorien osalta kaasumaisen polttoaineen moodissa.

([3]) Päästöjen raja-arvoja sovelletaan ainoastaan yli 70 prosentin kuormituksessa.

2 osa

Uusien keskisuurten polttolaitosten päästöjen raja-arvot

1. Muiden keskisuurten polttolaitosten kuin moottoreiden ja kaasuturbiinien päästöjen raja-arvot (mg/Nm^3)

Epäpuhtaus	Kiinteä biomassa	Muut kiinteät polttoaineet	Muut nestemäiset polttoaineet kuin raskas polttoöljy	Raskas polttoöljy	Maakaasu	Muut kaasumaiset polttoaineet kuin maakaasu
SO_2	200	400	170	350	-	35
NO_X	300	300	200	300	100	200
Hiukkaset	20([1])	20	20	20	-	-

([1]) 25 mg/Nm^3 niiden laitosten osalta, joiden lämpöteho on enintään 5 MW.

2. Moottoreiden ja kaasuturbiinien päästöjen raja-arvot (mg/Nm³)

Epäpuhtaus	Laitostyyppi	Nestemäiset polttoaineet	Maakaasu	Muut kaasumaiset polttoaineet kuin maakaasu
SO_2	Moottorit ja kaasuturbiinit	60	-	15
NO_X	Moottorit	190 ([1])	95 ([2])	190
	Kaasuturbiinit ([3])	75	50	75
Hiukkaset	Moottorit ja kaasuturbiinit	10	-	-

([1]) 225 mg/Nm³ kaksoispolttoainemoottorien osalta nestemäisen polttoaineen moodissa.

([2]) 190 mg/Nm³ kaksoispolttoainemoottorien osalta kaasumaisen polttoaineen moodissa.

([3]) Päästöjen raja-arvoja sovelletaan ainoastaan yli 70 prosentin kuormituksessa.

LIITE III

Vertailuarvot 5 artiklan 4 kohdassa tarkoitetuille tiukemmille päästöjen raja-arvoille

Kaikki tässä liitteessä vahvistetut päästöjen raja-arvot on määritettävä 273,15 K:n lämpötilassa ja 101,3 kPa:n paineessa ja savukaasujen vesihöyrypitoisuuden mukaan tehtävän korjauksen jälkeen sekä standardoituna happipitoisuuteen, joka on kiinteitä polttoaineita käyttävien polttolaitosten osalta 6 prosenttia, nestemäisiä ja kaasumaisia polttoaineita käyttävien muiden polttolaitosten kuin moottoreiden ja kaasuturbiinien osalta 3 prosenttia sekä moottoreiden ja kaasuturbiinien osalta 15 prosenttia.

Vertailuarvot muiden keskisuurten polttolaitosten kuin moottoreiden ja kaasuturbiinien päästöjen raja-arvoille (mg/Nm³)

Epäpuhtaus	Nimellinen lämpöteho (MW)	Kiinteä biomassa	Muut kiinteät polttoaineet	Nestemäiset polttoaineet	Maakaasu	Muut kaasumaiset polttoaineet kuin maakaasu
NO_X	1–5	200	100	120	70	120
	>5–50	145	100	120	70	120
Hiukkaset	1–5	10	10	10	-	-
	>5–50	5	5	5	-	-

Vertailuarvot moottoreiden ja kaasuturbiinien päästöjen raja-arvoille (mg/Nm³)

Epäpuhtaus	Laitostyyppi	Nestemäiset polttoaineet	Maakaasu	Muut kaasumaiset polttoaineet kuin maakaasu
NO_X	Moottorit	150	35	35
	Kaasuturbiinit ([1])	50	20	50

([1]) Vertailuarvoa sovelletaan ainoastaan yli 70 prosentin kuormituksessa.

LIITE IV

Päästöjen tarkkailu

1. Keskisuurissa polttolaitoksissa on tehtävä säännöllisin väliajoin SO_2-, NO_x- ja hiukkasmittauksia vähintään joka kolmas vuosi, kun laitoksen nimellinen lämpöteho on yli 1 MW ja alle 20 MW, ja vähintään kerran vuodessa, kun laitoksen nimellinen lämpöteho on 20 MW tai enemmän mutta alle 50 MW.

2. Mittauksia vaaditaan vain niistä epäpuhtauksista, joille on vahvistettu päästöjen raja-arvo liitteessä II asianomaisen laitoksen osalta.

3. Ensimmäiset mittaukset on tehtävä kolmen kuukauden kuluessa laitoksen rekisteröinnistä.

4. Edellä 1 kohdassa tarkoitettujen SO_2-mittausten sijaan SO_2-päästöt voidaan määrittää myös muilla toimivaltaisen viranomaisen todentamilla ja hyväksymillä menettelyillä.

5. Epäpuhtauksien näytteenotto ja analysointi sekä käyttöparametrien ja mahdollisten 4 kohdan nojalla käytettyjen vaihtoehtojen mittaukset on tehtävä CEN-standardien mukaisesti. Jos CEN-standardeja ei ole käytettävissä, on käytettävä ISO-, kansallisia tai muita kansainvälisiä standardeja, joilla varmistetaan vastaavaa tieteellistä tasoa olevat tiedot.