Escola Politécnica Superior

TRABAJO FIN DE MÁSTER

CURSO 2016/2017

ANTEPROYECTO PETROLERO DE 80.000 T.P.M.

Máster en Ingeniería Naval y Oceánica

CUADERNO VI

PREDICCIÓN DE POTENCIA Y DISEÑO DE PROPULSORES Y TIMONES

DEPARTAMENTO DE INGENIERÍA NAVAL Y OCEÁNICA

TRABAJO FIN DE MÁSTER

CURSO 2.016-2017

PROYECTO NÚMERO 17/27

TIPO DE BUQUE: Petrolero de crudo de 80.000 TPM

CLASIFICACIÓN, COTA Y REGLAMENTOS DE APLICACIÓN: LLOYD'S REGISTER OF SHIPPING. SOLAS. MARPOL. ILO. EXPANAMAX

CARACTERÍSTICAS DE LA CARGA: Transporte de petróleo crudo de densidad relativa 0,88. Calefacción de tanques.

VELOCIDAD Y AUTONOMÍA: 15 nudos en condiciones de servicio. 85 % MCR + 10% de margen de mar. 10.000 millas

SISTEMAS Y EQUIPOS DE CARGA / DESCARGA: Bombas de carga y descarga en cámara de bombas.

PROPULSIÓN: Diesel eléctrica con motores tipo dual fuel. Dos líneas de ejes con hélice de paso fijo.

TRIPULACIÓN Y PASAJE: 20 Personas en camarotes individuales.

OTROS EQUIPOS E INSTALACIONES: Los habituales en este tipo de buques.

Ferrol, Octubre de 2.016

ALUMNO: D. Jose Antonio González Llorente

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Capítulo 1. INTRODUCCIÓN

En este cuaderno se tratara de obtener la verdadera potencia propulsora que necesitara nuestro buque tras un estudio algo más exhaustivo que los realizados en cuadernos anteriores. Esto será posible, gracias al mayor conocimiento de nuestro buque (Formas, coeficientes etc.) ya que a medida que avanzamos en los cuadernos, vamos definiendo un mayor número de características al buque.

También se realizara como se indica en el título del cuaderno, un diseño de los propulsores y timones correspondientes al buque proyecto acompañado del cálculo de dimensiones correspondiente.

Los datos de partida de que se dispone para la realización de este cuaderno son los que se han venido determinando hasta ahora y que se resumen a continuación.

Características principales del buque	
Eslora entre perpendiculares (m)	220
Manga de trazado (m)	34
Puntal de trazado (m)	21
Calado de diseño (m)	15
Velocidad (nudos)	15
Coeficiente de bloque	0,842
Coeficiente en la maestra	0,996
Coeficiente prismático	0,846
Desplazamiento (Tn)	93492,4

Figura 1-1 - Características del buque
Fuente: Propia

Capítulo 2. ESTIMACIÓN DE LA POTENCIA PROP.

La potencia propulsora se puede determinar a partir de diferentes métodos, nosotros realizaremos el estudio de la estimación de la potencia propulsora de nuestro buque por el método de Holtrop, por ser un método muy efectivo y propio de grandes buques como petroleros.
Realizaremos el estudio de la potencia a instalar mediante métodos empíricos y el software Maxsurf Resistance, aunque siempre por el método de Holtrop.

Posteriormente cotejaremos los valores obtenidos y dependiendo de estos valores decidiremos cual será nuestra potencia propulsora a instalar.
Para poder diseñar el conjunto del sistema propulsivo del buque proyecto es básico e imprescindible conocer que potencia necesitamos. Para ello es necesario partir de la predicción de potencia del mismo, es decir, que para la realización de este cálculo se partirá de las características geométricas del buque (formas, compartimentado, etc.) y de todos los datos obtenidos a lo largo de cuadernos anteriores.

Luego la potencia propulsora estará delimitada por la velocidad exigida como un RPA. En este caso, la velocidad del buque deberá cumplir 15 nudos al calado de diseño con el motor al 85% de MCR y con un margen de mar del 10%.

2.1. POTENCIA PROPULSORA J MAU

Primera estimación Potencia propulsora J MAU

$$PB = 0{,}014 * V^3 * DWT^{0{,}55}$$

$$PB = 23.501{,}80\ Kw$$

2.2. MÉTODO DE D.G.M. WATSON

Aunque es una fórmula deducida especialmente para cargueros también puede aplicarse a petroleros y graneleros. Proporciona la potencia necesaria en condiciones de pruebas a plena carga con un margen de error aproximado del 10%.

$$PB = 0{,}889 * \Delta^{\frac{2}{3}} * \frac{(40 - \frac{Lpp}{61} + 400 * (k-1)^2 - 12 * Cb)}{15000 - 1{,}81 * N * Lpp^{0{,}5}} * Vp^3$$

Siendo:

- K= Constante de la fórmula de Alexander = 1,082
- V =Velocidad en nudos, en condiciones de pruebas a plena carga
- PB = Potencia desarrollada por el motor propulsor directamente acolado, en HP
- N = R.P.M. del motor propulsor = 105 rpm
- ∇=LPP*B*T*Cb=94472.4 Tn
- Δ=96.834,21 Tn
- $PB = 14.791{,}137\ HP$

Al ser un HP (horse power) igual a 0,745699872 kilovatios hacemos la conversión

$$PB = 14.791{,}137 * 0{,}746 = 11.029{,}75\ kW$$

2.3. CALCULO DE POTENCIA POR EL MÉTODO DE KUPRAS

El método presentado por L. K. Kupras parte del concepto de velocidad límite VB.

Es la velocidad por debajo de la cual el coeficiente de resistencia total no varía mucho y por encima de la cual comienza a aumentar rápidamente.

Esa velocidad varía en función del coeficiente de bloque y de la eslora según la fórmula:

$$MCO = PS = PB = \frac{PD * FS}{ETAM}$$

Velocidad limite

Partimos del concepto de velocidad límite VB, que es aquella por debajo de la cual el coeficiente de resistencia total no varía mucho y por encima empieza a aumentar rápidamente.

Esta velocidad es función del coeficiente de bloque y de la eslora.

$$VB = (3{,}08 - 2{,}54 * Cb) * LPP^{0{,}5}$$

$$VB = 13{,}962 \text{ Knots}$$

Potencia de remolque

$$PE = \frac{C * DISW^{\frac{2}{3}} * VB^3}{427{,}1}$$

Siendo C=0,71

9.526,72672

Potencia absorbida por la hélice a la velocidad limite

$$PDB = \frac{0{,}0023725 * (1 + x) * 0{,}71 * DISW^{\frac{2}{3}} * VB^3}{ETAD}$$

ETAD: rendimiento cuasi-propulsivo

$$ETAD = ETA0 * ETAH * ETAR$$

ETA0: rendimiento del propulsor en aguas libres

$$ETA0 = 1{,}3 - 0{,}55 * Cb - 0{,}00267 * N$$

$$ETA0 = 0{,}57791$$

ETAH: rendimiento del casco

Al ser nuestro coeficiente de bloque superior a 0,80 utilizaremos la siguiente expresión:

$$Cb = 0{,}842 > 0{,}8$$

$$ETAH = 0{,}945 + 0{,}11 * \frac{B}{T} + 20 * (Cb - 0{,}80) * (1{,}54 - (0{,}945 + 0{,}11 * \frac{B}{T}))$$

$$ETAH = 1{,}4847$$

ETAR: rendimiento rotativo relativo

$$\mathrm{ETAR} = 1{,}01$$

Teniendo todo calculamos ETAD

$$\mathrm{ETAD} = \mathrm{ETA0} * \mathrm{ETAH} * \mathrm{ETAR}$$

$$\mathrm{ETAD} = 0{,}57791 * 1{,}4847 * 1{,}01 = 0{,}8666$$

(1+x): factor de correlación

$$1 + \mathrm{x} = 0{,}85 + 0{,}00185 \left[\frac{1.000 - 3{,}28 * \mathrm{Lpp}}{100}\right]^{2{,}5}$$

$$1 + \mathrm{x} = 0{,}874$$

Potencia absorbida por la hélice a la velocidad limite

$$\mathrm{PDB} = \frac{0{,}0023725 * (1 + \mathrm{x}) * 0{,}71 * \mathrm{DISW}^{\frac{2}{3}} * \mathrm{VB}^{3}}{\mathrm{ETAD}}$$

$$\mathrm{PDB} = 9.735{,}746 \ \mathrm{HP}$$

$$\mathrm{PDB} = 7.259{,}9360 \ \mathrm{Kw}$$

Potencia para una velocidad distinta a la VB

$$\mathrm{PD} = \mathrm{PDB} * \left(\frac{\mathrm{V}}{\mathrm{VB}}\right)^{4{,}167 * \frac{\mathrm{V}}{\mathrm{B}}}$$

$$\frac{\mathrm{V}}{\mathrm{VB}} = 1{,}0743$$

$$\mathrm{PD} = 11.107{,}674 \ \mathrm{HP}$$

$$\mathrm{PD} = 8.282{,}981 \ \mathrm{Kw}$$

Potencia desarrollada por el motor propulsor a la velocidad de proyecto será:

$$\mathrm{MCO} = \mathrm{PS} = \mathrm{PB} = \frac{\mathrm{PD} * \mathrm{FS}}{\mathrm{ETAM}} = 11.334{,}361 \ \mathrm{HP} = 8.452{,}022 \ \mathrm{Kw}$$

Siendo el factor de servicio (FS)=1

Siendo el rendimiento mecánico (ETAM) para buques de una hélice, motores directamente acoplados y cámara de máquinas a popa=0,98

2.4. TABLA DE REGRESIÓN DE ESTUDIO ESTADÍSTICO

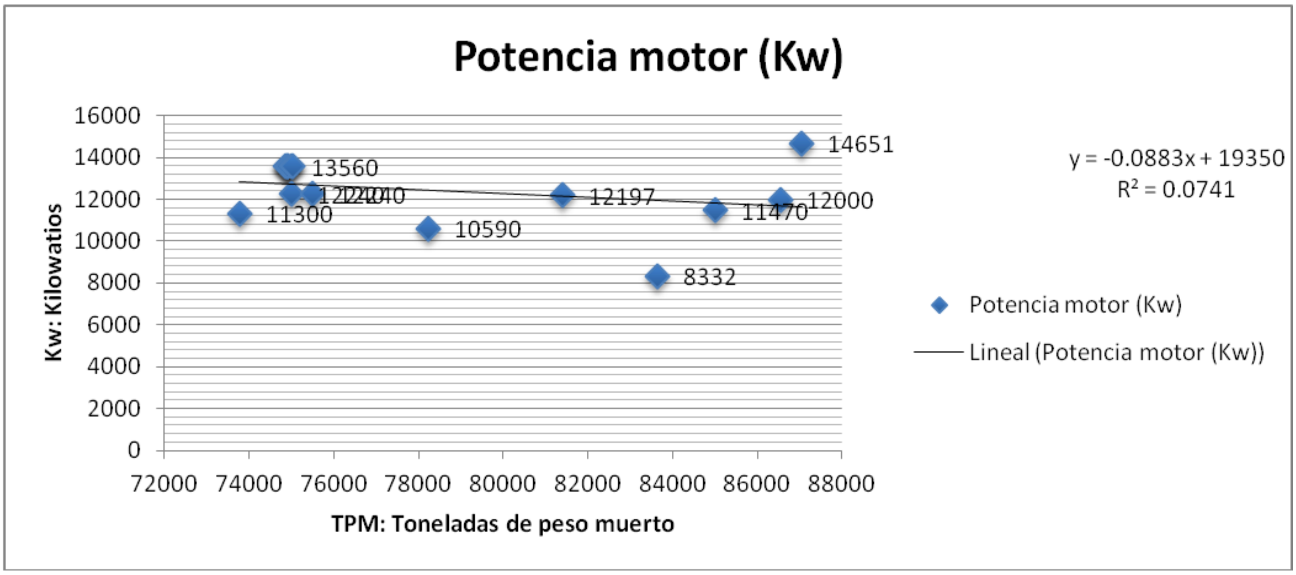


Figura 2-1 - Tabla de regresión potencias

Fuente: Propia

$$Potencia\ (Kw) = 0{,}0883 * 80.000 + 19350 = 12.286\ Kw$$

Para una estimación de la resistencia y la motorización de nuestro buque, se ha empleado el software Resistance, en el que con las formas de nuestro buque como input, así como, las velocidades que requeremos (de 0 a 15 nudos), nos estima la resistencia y potencia que necesitaríamos.

Hemos estimado la eficiencia del motor en un 85% para tener un coeficiente de seguridad.

Una vez calculada la resistencia total, la potencia de remolque se obtiene con la siguiente expresión:

$$EHP = \frac{Rt * V}{75}$$

2.5. ESTIMACIÓN DE LA POTENCIA MEDIANTE EL SOFTWARE MAXSURF RESISTANCE

Hay varios métodos para poder estimar la potencia propulsora necesaria para mover nuestro buque a 15 nudos de velocidad de crucero, nosotros procederemos a calcular la potencia propulsora con el Método de Holtrop, ya que he observado consultando otros proyectos de buques petroleros de crudo de porte parecido, que es un método muy utilizado para el cálculo de la potencia de este tipo de buques.

Para realizar la estimación por el Método de Holtrop de la potencia necesaria para mover nuestro buque al calado de proyecto y a la velocidad de servicio, nos ayudamos del software Resistance de Maxsurf.

Procedemos con el software Resistance a importar las formas de nuestro buque definidas previamente con el software Maxsurf, y con él lo predefinimos para que nos realice los cálculos por el Método de Holtrop.

Un aspecto muy importante que hemos tenido que calcular para nuestro y buque y posteriormente introducir en el Resistance es el Rendimiento de la Propulsión, es decir, esto es el rendimiento de la instalación completa, desde la que producimos en la planta propulsora, pasando por la que llega a la hélice siendo aprovechada por ella, y hasta la que finalmente se emplea en vencer la resistencia al avance del buque.

Este rendimiento de la instalación propulsora completa lo acabamos de definir en el apartado siguiente, que como sabemos tiene un valor del 53%, así que ya podemos introducir los datos, preparar el software y realizar la estimación de potencia con el Resistance.

Es totalmente necesario introducir dicho rendimiento, debido a que el software Maxsurf por defecto nos calcula la potencia expresada como EHP, como lo que nosotros buscamos son los BHP necesitamos conocer el rendimiento de la instalación propulsora completa.

Veamos ahora una tabla extraída directamente del software Resistance, con los parámetros de entrada necesarios para la realización de los cálculos. En dicha tabla se puede observar los valores extraídos de las formas de nuestro buque que usará el Resistance para la aplicación del método de Holtrop:

	Item	Value	Units	Holtrop
1	LWL	226,174	m	226,174
2	Beam	33,999	m	33,999
3	Draft	15,002	m	15,002
4	Displaced volume	94442,95	m^3	94442,95
5	Wetted area	12205,672	m^2	12205,67
6	Prismatic coeff. (Cp)	0,845		0,845
7	Waterpl. area coeff. (Cwp)	0,915		0,915
8	1/2 angle of entrance	26,5	deg.	26,5
9	LCG from midships(+ve for'	6,089	m	6,089
10	Transom area	2,89	m^2	2,89
11	Transom wl beam	2,567	m	--
12	Transom draft	2,152	m	--
13	Max sectional area	508,06	m^2	--
14	Bulb transverse area	55,279	m^2	55,279
15	Bulb height from keel	9,937	m	9,937
16	Draft at FP	15	m	15
17	Deadrise at 50% LWL	0	deg.	--
18	Hard chine or Round bilge	Round bilge		--
19				
20	Frontal Area	0	m^2	
21	Headwind	0	kn	
22	Drag Coefficient	0		
23	Air density	0,001	tonne/	
24	Appendage Area	0	m^2	
25	Nominal App. length	0	m	
26	Appendage Factor	1		
27				
28	Correlation allow.	0,0004		Calculate
29	Kinematic viscosity	0,0000011	m^2/s	
30	Water Density	1,026	tonne/	

Figura 2-2 - Datos de entrada para el cálculo de potencia

Fuente: Maxsurf Resistance

Como podemos observar, los valores de los coeficientes son perfectos para este método, de lo contrario aparecerían en color naranja o rojo indicativo de que el valor en cuestión sería elevado para la realización de la potencia por este método. Por lo que podemos comprobar todos los demás parámetros están dentro de los ideales para la aplicación de este método, por lo que como ya dijimos anteriormente el método de Holtrop el método más conveniente para el estudio de la resistencia de un petrolero.

Una vez definidas las formas del buque, es posible deducir la potencia y/o empuje necesarios para la propulsión efectiva del buque. Existen varios métodos de predicción, como Holtrop, Series 60, Van Oortmeersen, etc. y en este caso se aplicara el desarrollado por el primero y el segundo.

Los resultados obtenidos se generaron mediante el software “Resistance”, pues trabaja directamente con las formas generadas mediante “Maxsurf”, y presentadas en el cuadernillo 3. Dicho programa genera una cubica de velocidad en función del empuje (T) y potencia (EHP) respectivamente. Se marcó el punto de velocidad 15 nudos dentro de las gráficas y la tabla de datos.

Los datos implicados son:

Se ha estimado mediante los métodos de Holtrop y de las series 60.

Speed (kts)	Holtrop Resist (kN)	Holtrop power (kW)	Series 60 Resist. (kN)	Series 60 Power (kW)
0	--	--	--	--
0.375	0.82	0.18	--	--
0.75	2.99	1.28	--	--
1.125	6.42	4.13	--	--
1.5	11.05	9.48	--	--
1.875	16.88	18.09	--	--
2.25	23.88	30.71	--	--
2.625	32.04	48.08	--	--
3	41.37	70.94	--	--
3.375	51.84	100	--	--
3.75	63.45	136	--	--
4.125	76.19	179.63	--	--
4.5	90.05	231.62	--	--
4.875	105.02	292.63	--	--

Speed (kts)	Holtrop Resist (kN)	Holtrop power (kW)	Series 60 Resist. (kN)	Series 60 Power (kW)
5.25	121.08	363.36	--	--
5.625	138.24	444.47	--	--
6	156.46	536.6	--	--
6.375	175.74	640.41	--	--
6.75	196.07	756.5	--	--
7.125	217.43	885.51	--	--
7.5	239.8	1028.02	--	--
7.875	263.17	1184.64	--	--
8.25	287.54	1355.97	--	--
8.625	312.9	1542.6	--	--
9	339.24	1745.18	--	--
9.375	366.66	1964.85	--	--
9.75	395.31	2203.1	--	--
10.125	425.03	2459.83	--	--
10.5	455.85	2735.96	--	--
10.875	487.84	3032.51	--	--
11.25	521.05	3350.65	--	--
11.625	555.57	3691.74	650.9	4325.16
12	591.51	4057.33	698.54	4791.5
12.375	628.99	4449.21	746.19	5278.26
12.75	668.15	4869.46	793.8	5785.2
13.125	709.17	5320.43	838.33	6289.38
13.5	752.24	5804.82	882.85	6812.65
13.875	797.59	6325.68	927.37	7355.01
14.25	845.44	6886.43	983.72	8012.73

Speed (kts)	Holtrop Resist (kN)	Holtrop power (kW)	Series 60 Resist. (kN)	Series 60 Power (kW)
14.625	896.07	7490.93	1041.07	8703.07
15	949.77	8143.41	1098.43	9418.01

Tabla 2-1 - Salida Resistance potencias

Fuente: Propia

2.5.1. POTENCIA VS VELOCIDAD

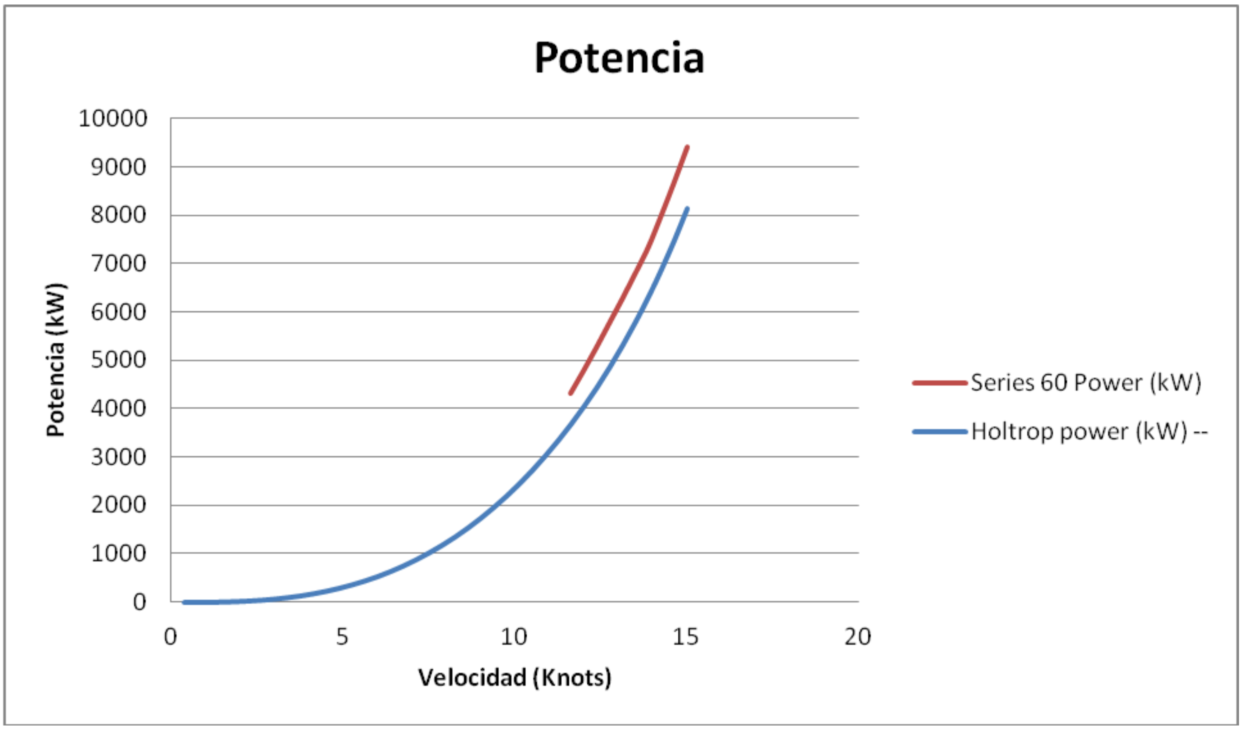


Figura 2-3 - Grafica potencia-velocidad

Fuente: Propia

2.5.2. Coeficiente Resistencia total Vs velocidad

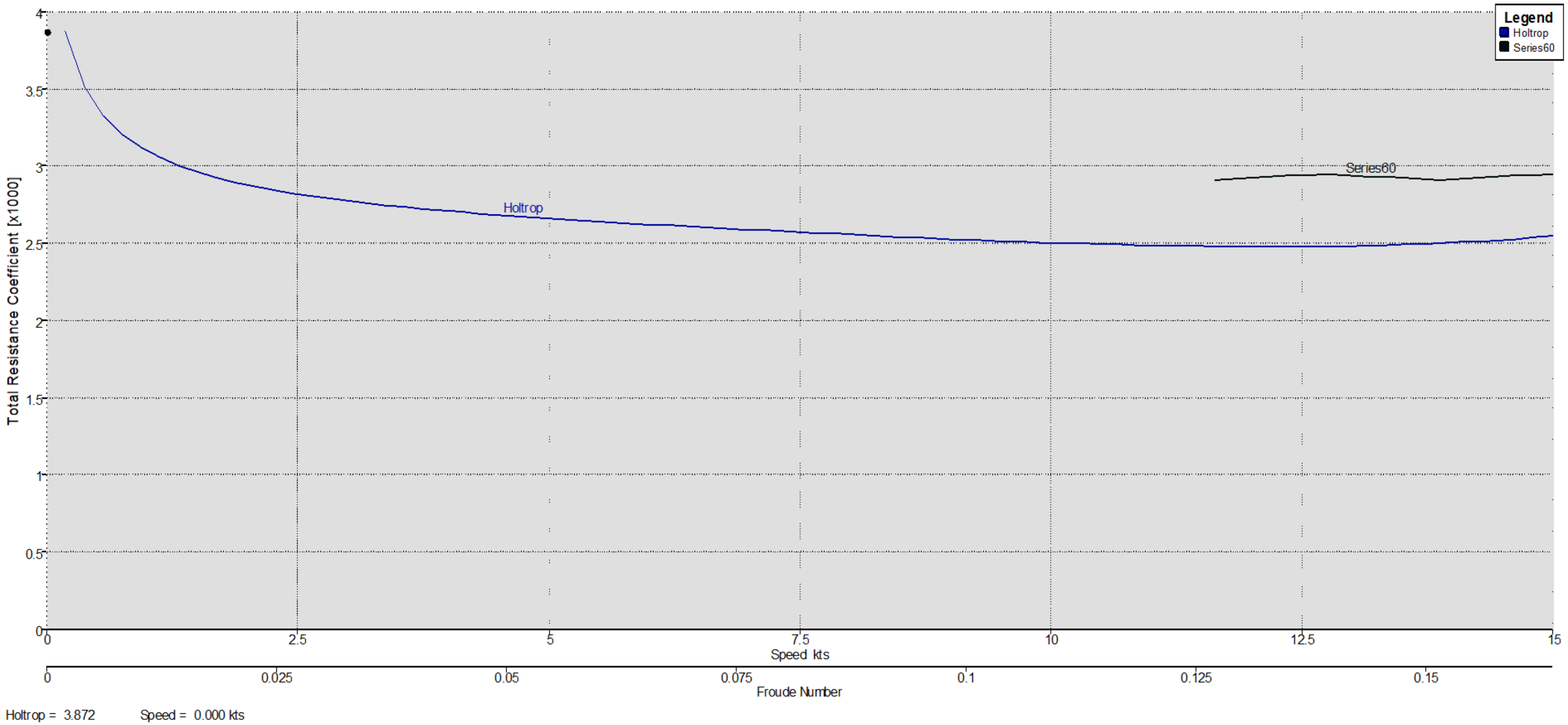


Figura 2-4 - Grafica Coeficiente resistencia total-velocidad

Fuente: Resistance

2.5.3. Coeficiente Resistencia residual Vs Velocidad

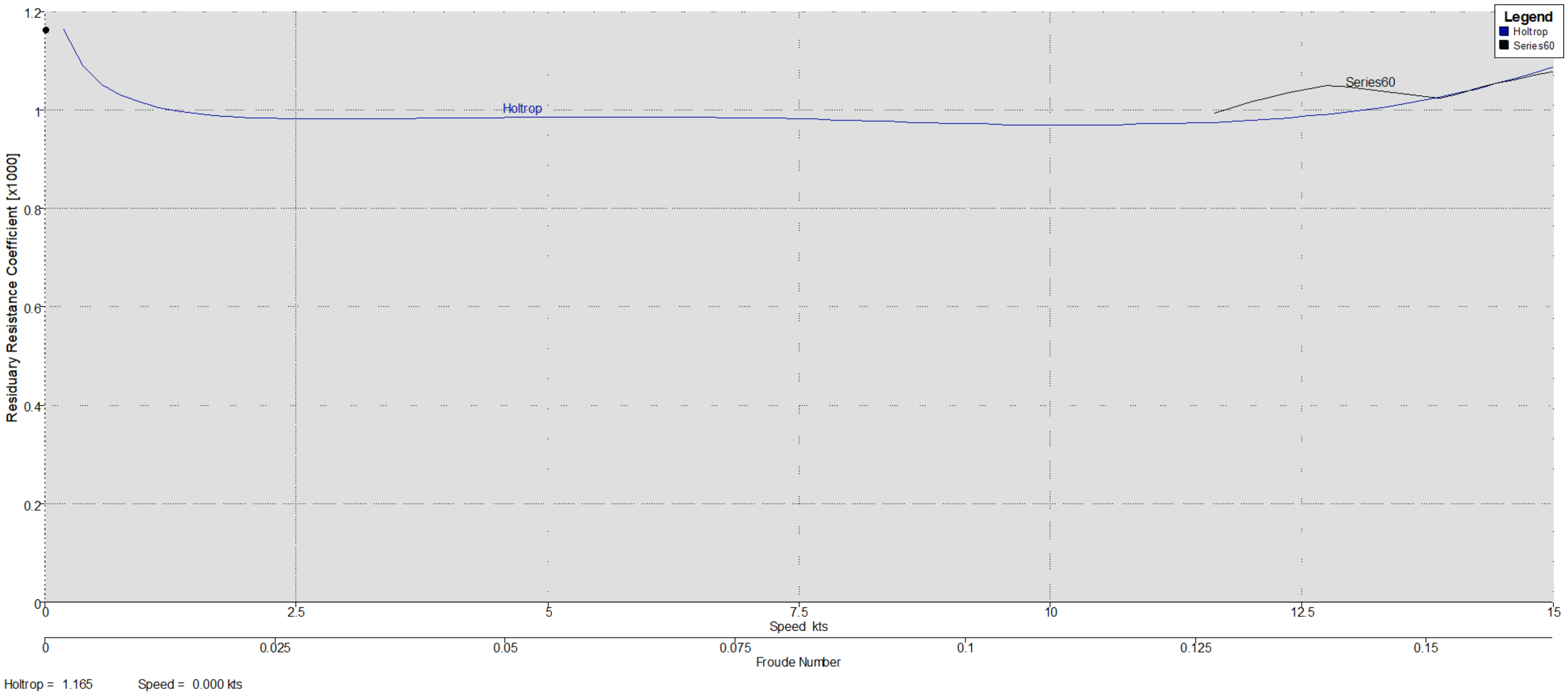


Figura 2-5 - Grafica Coeficiente resistencia residual-velocidad
Fuente: Resistance

2.5.4. Coeficiente Resistencia por formación de olas Vs Velocidad

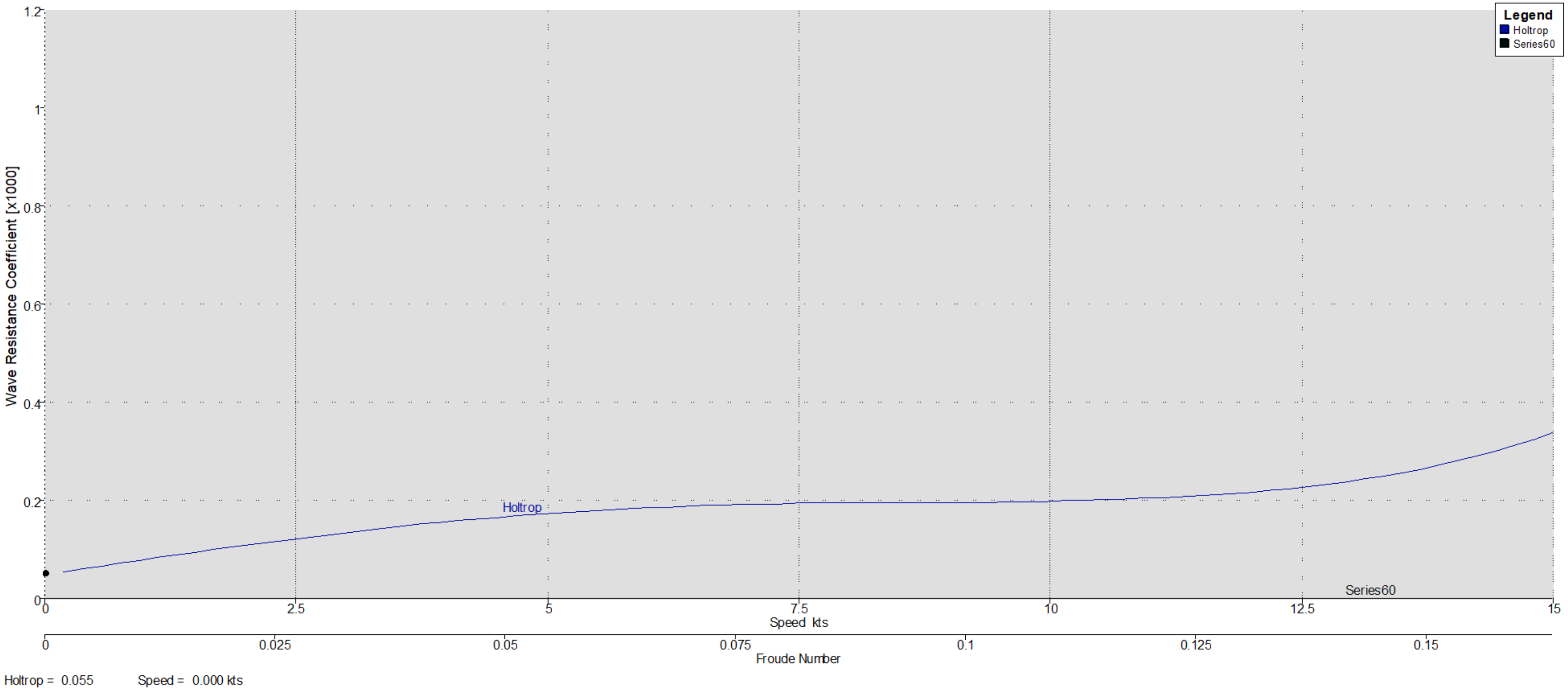


Figura 2-6 - Grafica Coeficiente resistencia por formación de olas-velocidad

Fuente: Resistance

2.5.5. Coeficiente Resistencia de fricción Vs Velocidad

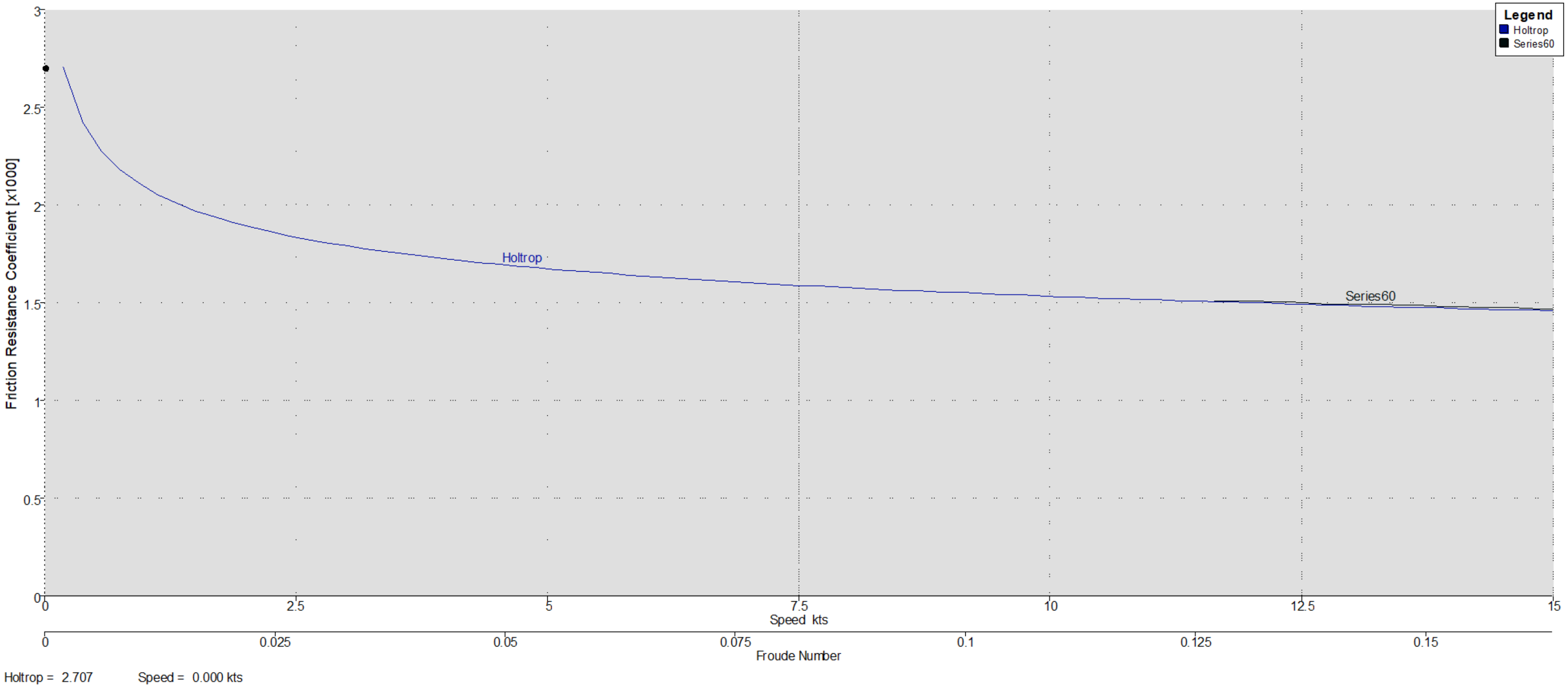


Figura 2-7 - Grafica Coeficiente resistencia de fricción-velocidad

Fuente: Resistance

2.5.6. COEFICIENTE RESISTENCIA VISCOSA VS VELOCIDAD

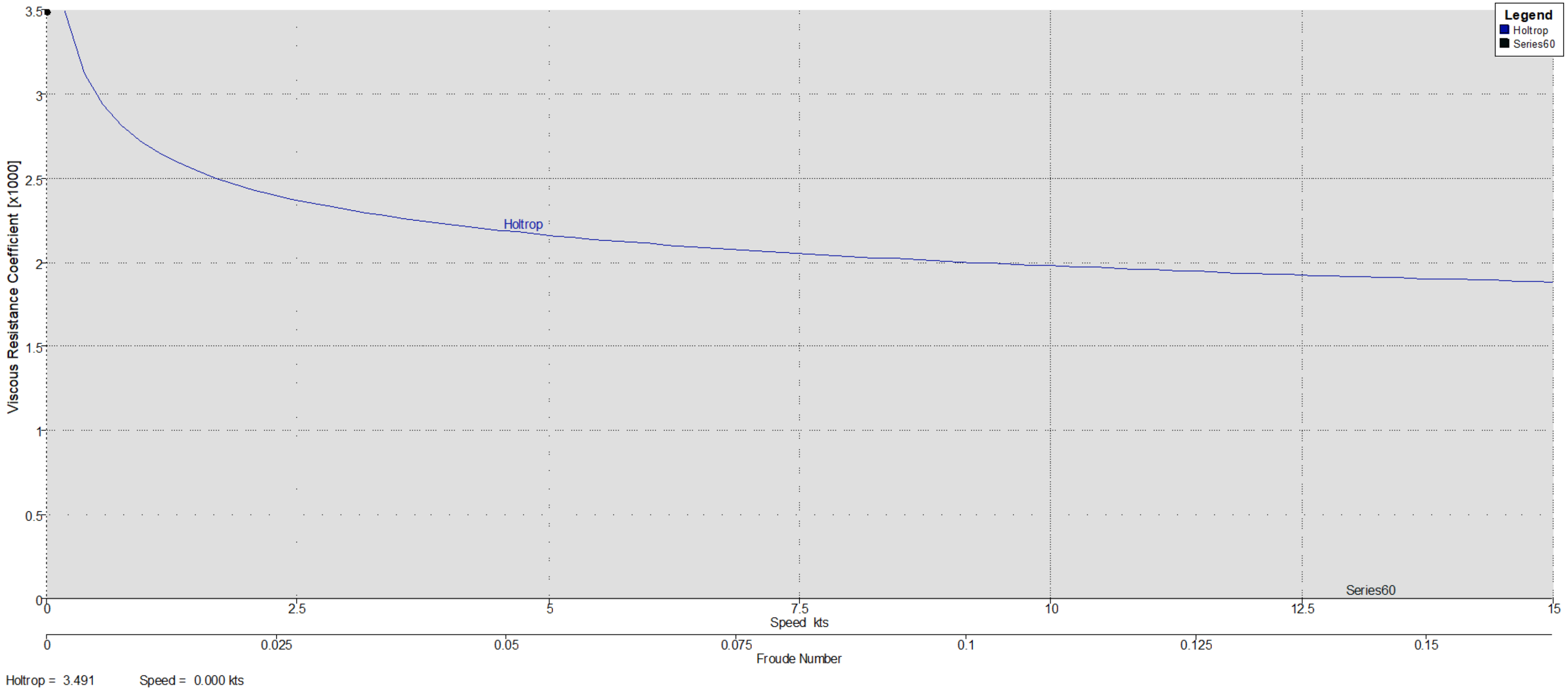


Figura 2-8 - Grafica Coeficiente resistencia viscosa-velocidad

Fuente: Resistance

Resumen de las potencias:

Resumen	Potencia (kW)
D.G.M Watson	11.029,749
Holtrop (Hullspeed)	8.143,41
J MAU	23.501,8
Kupras	8.542,022
Series 60	9.418,01
Tabla regresión	12.286

Tabla 2-2 - Resumen potencias
Fuente: Propia

2.6. POTENCIA DE SALIDA DEL MOTOR PRINCIPAL

Desde que la potencia es generada por la planta de propulsión hasta que se transforma en una fuerza de empuje necesaria para mover el buque en las hélices, la potencia realiza un "recorrido", por el que evidentemente se va perdiendo potencia, es decir, como la potencia desde que es generada hasta que es útil se va perdiendo progresivamente por el camino dependiendo donde la midamos nos dará un valor u otro, esto es lógico ya que el rendimiento de los distintos elementos en la cadena de transmisión no será nunca del 100% debido a los rozamientos existentes entre sus partes. Es por esto que denominamos de manera distinta la potencia medida en un sitio que en otro, para saber en qué punto está la potencia a la que nos referimos.

Esta es la forma habitual de referirnos a la potencia, a lo largo de su "recorrido":

La evaluación de la potencia que deberá desarrollar el motor propulsor se lleva a cabo a través de su relación con la potencia efectiva y los distintos rendimientos, es decir:

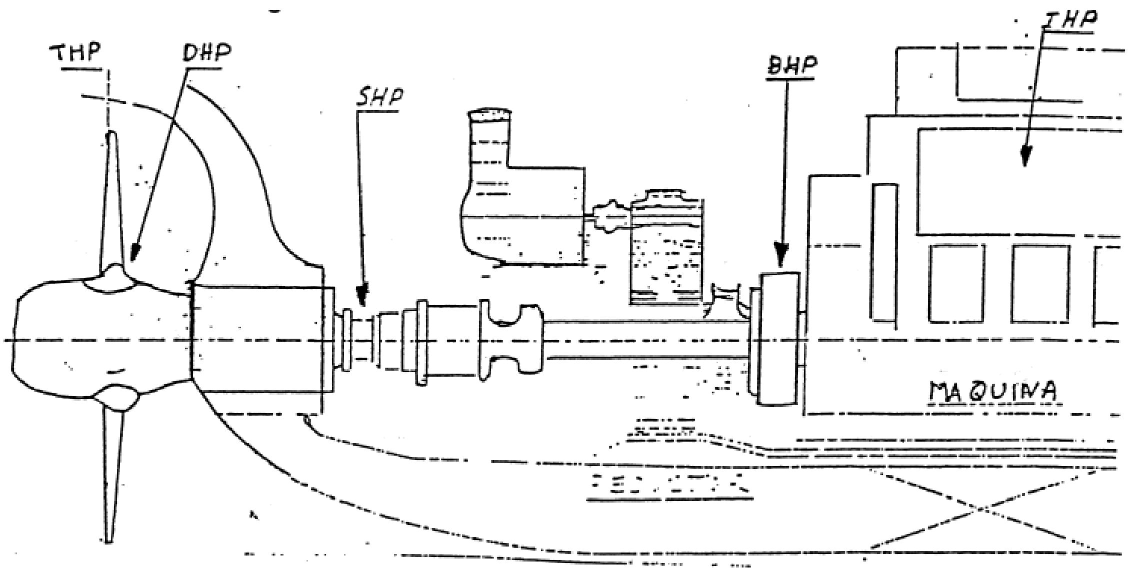


Figura 2-9 - Tipos de potencia

Fuente: Apuntes fundamentos de teoría del buque

Explicaremos brevemente y de forma resumida en que consiste cada tipo de potencia y finalmente podremos calcular el Rendimiento de la instalación propulsora completa, también llamado como Coeficiente de propulsión:

- **Potencia indicada IHP (Indicated Horse Power)**
Es la potencia teórica determinada por el volumen y la presión de los cilindros, en la realidad los motores nunca dan la potencia teórica, debido a que los ciclos térmicos que en ellos se dan no son ideales.
- **Potencia al freno BHP (Brake Horse Power)**
Es la potencia medida en el plato de acoplamiento del cigüeñal al eje mediante la aplicación de un par de frenado, es decir es la potencia real que nos pueda dar el motor, debido a lo que dijimos antes, ya que lo motores no son ideales, no pueden dar toda la potencia que teóricamente deberían por lo que la verdadera potencia que son capaces de generar es la BHP, ya que es la medida en su salida al eje.
- **Potencia en el eje SHP (Shaft Horse Power)**
Es la potencia medida en la parte del eje más próxima a la bocina, es decir en la parte última del eje, y por lo tanto vemos como se ha perdido la potencia durante dicho eje, desde que en él entraron los “BHP”, hasta que por él salen “SHP”.
- **Potencia en el propulsor DHP (Delivered Horse Power)**
Es la potencia que recibe directamente la hélice, no es medible pero se puede calcular como la potencia medida en el eje SHP menos las perdidas en la bocina, que se pueden estimar del orden de un 3% aproximadamente.
- **Potencia de empuje THP (Thrust Horse Power)**
Es la potencia desarrollada por el empuje realizado por la hélice, debido a que el agua que llega a la hélice va a una velocidad (llamada velocidad de avance) inferior a la del buque debido al efecto de arrastre del agua por la carena, habrá una pérdida de potencia en la hélice, de manera que la hélice tiene también un rendimiento que no es del 100%.

- **Potencia efectiva EHP (Efective Horse Power)**
Es la empleada en vencer la resistencia al avance del buque a una determinada velocidad. Se puede calcular como la potencia de empuje THP multiplicada por el rendimiento del casco y por el coeficiente de colocación de la hélice.
Una vez dicho esto, procederemos al cálculo del rendimiento de la instalación propulsora completa, es decir, a la relación entre la potencia aprovechada para mover el buque EHP y la potencia producida en la planta propulsora de forma teórica IHP.

Rendimiento de la instalación propulsora o Coeficiente de la propulsión:

$$Coeficiente\ de\ propulsión: \frac{EHP}{IHP}$$

Rendimiento de la instalación propulsora:

$$\frac{EHP}{IHP} = \eta = \frac{EHP}{DHP} * \frac{DHP}{BHP} * \frac{BHP}{IHP}$$

Así que una vez dicho esto denominaremos:

$$Rendimiento\ propulsivo: \frac{EHP}{DHP}$$

$$Rendimiento\ de\ las\ transmisiones: \frac{DHP}{BHP}$$

$$Rendimiento\ mecánico: \frac{BHP}{IHP}$$

Como lo que yo voy a buscar cuando vaya a determinar el motor a instalar es la potencia real que es capaz de dar, es decir, los BHP del motor; para calcular el rendimiento de la instalación propulsora, no tendremos en cuenta el rendimiento mecánico, ya que eso es cosa de los fabricantes, nosotros directamente buscaremos un motor por sus BHP, de manera que la expresión anterior nos queda de la siguiente forma:

$$\frac{EHP}{BHP} = \eta = \frac{EHP}{DHP} * \frac{DHP}{BHP}$$

Despejando las BHP, que es el dato que nos interesa:

$$BHP = \frac{EHP}{\eta prop}$$

Desglosando el rendimiento propulsivo en los distintos rendimientos no queda:

$$BHP = \frac{EHP}{\eta p} = \frac{EHP}{\eta H * \eta cc * \eta m * \eta 0}$$

El rendimiento de las transmisiones se compone de las pérdidas en los cojinetes del eje, chumacera de empuje, perdidas en bocina y pérdidas en reductoras.

Es decir que el rendimiento propulsivo es el producto del rendimiento de la hélice por el rendimiento del casco y por el coeficiente de colocación de la hélice.

El rendimiento del casco depende de las formas de popa del buque, obteniéndose su valor mediante ensayos de autopropulsión con modelos.

El coeficiente de colocación de la hélice depende de la cantidad de flujo laminar y turbulento que incide sobre las palas de la hélice y de la colocación de la misma.

Siendo:

- BHP: potencia de freno o de salida del motor
- EHP: potencia efectiva o de remolque
- ηp: rendimiento del propulsor
- ηH: rendimiento del casco
- ηrr: rendimiento rotativo-relativo
- ηm: rendimiento mecánico
- $\eta 0$: rendimiento del propulsor en aguas libres

Estos componentes se estiman a partir de los requerimientos de proyecto, diversas fórmulas empíricas y las series sistemáticas de propulsores.

2.7. CÁLCULO DE LOS RENDIMIENTOS

A continuación, vamos a realizar el cálculo estimativo de los rendimientos propulsivos del propulsor en aguas libres manualmente, para obtener una estimación de los valores finales que obtendremos mediante el software Navcad.

2.7.1. RENDIMIENTO DEL CASCO

El rendimiento del casco depende de las formas de popa del buque, obteniéndose su valor mediante ensayos de autopropulsión con modelos. Su valor oscila normalmente entre 0,9 y 1,1, siendo los más cercanos a 1,1 los buques mejor proyectados. Nosotros tomaremos el valor medio para esta estimación, consiguiendo así un valor promedio más acertado para iniciar los cálculos, es decir, tomaremos un valor del rendimiento del casco con valor la unidad.

2.7.2. RENDIMIENTO MECÁNICO

Para buques de dos hélices, motores directamente acoplados y cámara de máquinas a popa resulta un valor de 0,95 en la condición de pruebas.

2.7.3. RENDIMIENTO DEL PROPULSOR EN AGUAS LIBRES

El rendimiento del propulsor oscila en torno a 0,55, sin embargo desconocemos este rendimiento por lo tanto vamos a estimar un valor menor a este para conocer la potencia mínima que deberemos obtener en un estudio más detallado de la predicción de potencia.

$$\eta_0 = 0{,}5$$

2.7.4. RENDIMIENTO PROPULSIVO

El rendimiento propulsivo es el producto del rendimiento de la hélice por el rendimiento del casco y por el coeficiente de colocación de la hélice.

$$\eta_{prop} = \eta H * \eta rr * \eta m * \eta_0$$

$$\eta_{prop} = 1 * 0{,}71 * 0{,}95 * 0{,}5 = 0{,}338$$

2.7.5. POTENCIA AL FRENO Y VELOCIDAD ESTIMADA

El coeficiente w representa el efecto de la acción de la carena sobre la hélice.

- t (coeficiente de succión):0,26
- w (coeficiente de estela):0,43
- ηH: rendimiento del casco:1
- ηrr: rendimiento rotativo-relativo:0.71
- ηm: rendimiento mecanico:0,95
- η0: rendimiento del propulsor en aguas libres:0,5

Estos componentes se estiman a partir de los requerimientos de proyecto, diversas fórmulas empíricas y las series sistemáticas de propulsores.

Por lo que obtenemos que:

$$BHP = 13.858{,}44\ kW$$

A continuación calcularemos la potencia del motor, ya que la anterior es la potencia al freno.

Aplicándole un coeficiente del 85% de trabajo de motor, obtenemos una potencia de:

$$MCR = 16.304{,}04\ kW$$

Para alcanzar la velocidad máxima usaremos el 90% de potencia del motor, es decir:

$$MCR90\% = 14.673.64$$

A continuación se refleja en la tabla la gráfica de potencias vs kW:

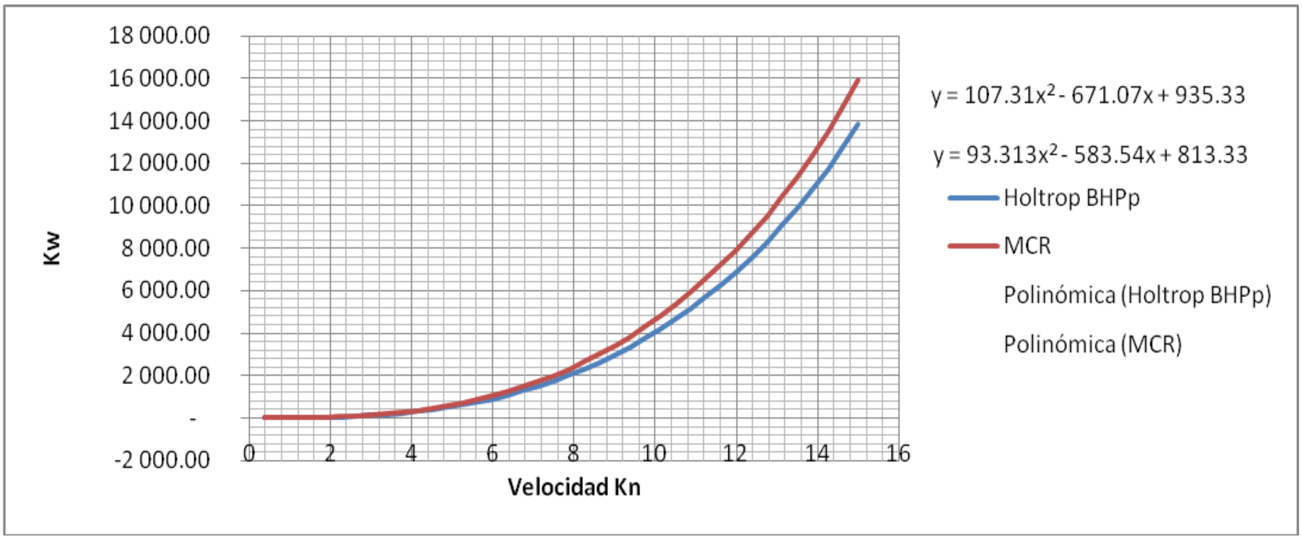


Figura 2-10 - Grafica potencia del motor-Holtrop

Fuente: Propia

Aplicando los factores de regresión, obtenemos que la velocidad de servicio para esa potencia será de:

$$y = 107{,}31x^2 - 671{,}07x + 935{,}33$$

a	107,310
b	- 671,070
c	- 15.368,714

Tabla 2-3 - Variables tabla de regresión Holtrop-Velocidad de servicio

Fuente: Propia

$$Vservicio = 15{,}50\ Knots$$

Mientras que aplicando el factor de la velocidad máxima obtendremos:

$$y = 93{,}313x^2 - 583{,}54x + 813{,}33$$

a	93,313
b	- 583,540
c	- 13.860,31

Tabla 2-4 - Variables tabla de regresión Holtrop-Velocidad máxima

Fuente: Propia

$$Vmaxima = 15{,}71\ Knots$$

Observamos que cumplimos perfectamente la velocidad de servicio del buque, la cual eran 15 nudos y la velocidad máxima de 15,7 nudos. Podríamos incluso forzar menos el motor para bajar a 15 nudos.

Cabe destacar que nuestro buque empleará la práctica del Slow steaming para sus rutas, donde ahorrará combustible.

Capítulo 3. ESTIMACIÓN DEL DIÁMETRO DE LA HÉLICE

Tal y como está descrito en las especificaciones, tendremos dos líneas de ejes, cada una de estas líneas de ejes dispondrá de una hélice única y de paso fijo.

Para el estudio del propulsor, se han de tener presente dos ideas fundamentales:

- La hélice debe proporcionar al buque el empuje necesario para que este pueda navegar a la velocidad que se especifique. Esto se debe conseguir con un rendimiento máximo, es decir, de manera que la potencia absorbida sea mínima. En el caso de este proyecto se tiene que cumplir, que en condiciones de pruebas de mar, el buque debe alcanzar una velocidad de 15 nudos al 85% de la potencia máxima continua (MCR).
- No se deben presentar fenómenos de cavitación, y en caso de que aparezcan, deben quedar reducidos a límites admisibles.

3.5.4 - ESTIMACIÓN DEL DIÁMETRO DE LA HÉLICE PROPULSORA

Es conveniente hacer una estimación del diámetro de la(s) hélice(s), DP, que permita, entre otras cosas, controlar su inmersión en las situaciones de navegación en lastre, y verificar los huelgos entre la misma y el casco del buque, que tienen una gran incidencia sobre aspectos muy importantes, como las vibraciones excitadas por la hélice.

La fórmula siguiente calcula el diámetro en metros de una hélice de palas fijas, a partir únicamente de la potencia del equipo propulsor y de las RPM de la hélice. También puede utilizarse la misma fórmula para estimar el diámetro de las hélices de paso controlable, que no difiere sensiblemente de las de palas fijas. No considera otros factores, como el número de palas, que también influyen sobre el diámetro.

$$DP = 15{,}75\,\frac{MCO^{0,2}}{N^{0,6}} \qquad \textbf{(3.5.37)}$$

Figura 3-1 - Estimación de la hélice propulsora

Fuente: El proyecto básico del buque mercante

a) Buques de 1 hélice

$a = K_z \times K \times DP$	(con un mínimo de 0,10 DP)	**(3.5.38)**
$b = 1{,}5\ a$	(id 0,15 DP)	**(3.5.39)**
$c = 0{,}12\ DP$	(id TMAX)	**(3.5.40)**
$d = 0{,}03\ DP$		**(3.5.41)**

b) Buques de 2 hélices:

$e = a$	(id de 0,20 DP si Z=3 ó 4)	**(3.5.42)**
	(id de 0,16 DP si Z=5 ó 6)	
$f = a$	(id de 0,15 DP)	**(3.5.43)**

Figura 3-2 - Estimación de la hélice propulsora, tipo de buque

Fuente: El proyecto básico del buque mercante

Para el primer cálculo aproximativo utilizaremos la siguiente expresión.

$$DP = 15{,}75^{\frac{MCO^{0,2}}{N^{0,6}}}$$

$$DP = 6{,}915\ m$$

Otro método seria mediante la entrada con los BHP de nuestro motor en la siguiente gráfica.

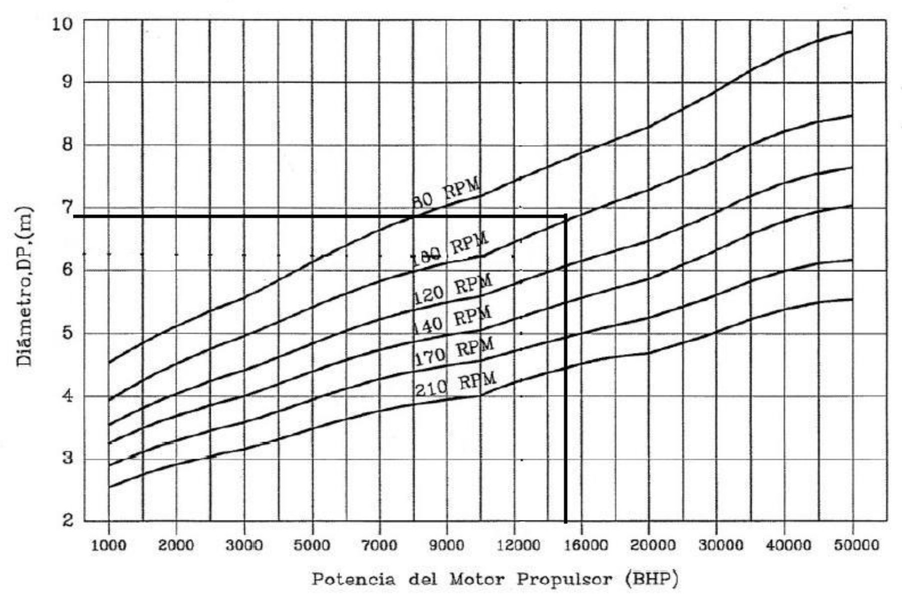


Figura 3-3 - grafica BHP-RPM

Fuente: El proyecto básico del buque mercante

Siendo nuestras BHP de 13.858,44 y teniendo nuestro motor 540 RPM vemos que el valor en la gráfica se ajusta al del método anterior, por lo que:

$$DP = 7\ metros$$

Capítulo 4. Predicción de potencia

Una vez realizada la estimación de la potencia necesaria, realizaremos el dimensionamiento propulsor de la siguiente manera:

1. Realizaremos una predicción de potencia mediante el software NavCAD por el método “By thrust” para calcular la potencia al freno (BHP) que necesitaremos para alcanzar la velocidad de diseño (15 nudos). Para ello utilizaremos los datos de la hélice del buque base.
2. Una vez obtengamos como dato de salida del paso 1 las BHP, entraremos a dimensionar el diámetro de la hélice.
3. Con el diámetro definido de nuestra hélice, volvemos a realizar la predicción de potencia mediante el software NavCAD por el método “By thrust” para calcular la potencia al freno (BHP) para 15 nudos de velocidad.
4. Con el resultado de la potencia que necesitamos, elegiremos los motores eléctricos idóneos para generar la potencia propulsiva.

5. Elegidos los motores, realizaremos la predicción de potencia, esta vez mediante el método “By power” ya que tenemos los datos del motor eléctrico elegido.

4.1. Estimación de potencia mediante hélice buque base

Realizamos el estudio de potencia con el diámetro de la hélice de nuestro buque base, de 8500 mm.

Propulsion
3 Jun 2017 12:08
HydroComp NavCad 2014

Project ID: **Petrolero 80.000 TPM**
Description: **Cuaderno 1**
File name: **Cuaderno 1.hcnc**

Analysis parameters

Hull-propulsor interaction		
Technique:	[Calc]	Prediction
Prediction:		Holtrop
Reference ship:		
Max prop diam:		8500,0 mm
Corrections		
Viscous scale corr:	[On]	Custom
Rudder location:		Behind propeller
Friction line:		ITTC-57
Hull form factor:		1,327
Corr allowance:		0,000163
Roughness [mm]:	[Off]	0,00
Ducted prop corr:	[Off]	
Tunnel stern corr:	[Off]	
Effective diam:		
Recess depth:		

System analysis	
Cavitation criteria:	Keller eqn
Analysis type:	Free run
CPP method:	
Engine RPM:	
Mass multiplier:	
RPM constraint:	
Limit [RPM/s]:	
Water properties	
Water type:	Salt
Density:	1026,00 kg/m3
Viscosity:	1,18920e-6 m2/s

Prediction method check [Holtrop]

Parameters	FN [design]	CP	LWL/BWL	BWL/T
Value	0,16	0,85	6,65	2,27
Range	0,06--0,80	0,55--0,85	3,90--14,90	2,10--4,00

Prediction results [System]

	HULL-PROPULSOR				ENGINE			
SPEED [kt]	PETOTAL [kW]	WFT	THD	EFFR	RPMENG [RPM]	PBPROP [kW]	FUEL [L/h]	LOADENG [%]
5,00 !	344,9	0,1941	0,2027	1,0293	31	345,4	—	0,0
7,00	902,5	0,1933	0,2027	1,0293	43	910,8	—	0,0
9,00	1850,7	0,1928	0,2027	1,0293	55	1879,5	—	0,0
10,00	2506,8	0,1926	0,2027	1,0293	61	2551,8	—	0,0
11,00	3315,7	0,1924	0,2027	1,0293	67	3379,8	—	0,0
12,00	4321,3	0,1922	0,2027	1,0293	73	4403,3	—	0,0
13,00	5596,4	0,1920	0,2027	1,0293	80	5687,5	—	0,0
14,00	7255,2	0,1919	0,2027	1,0293	86	7334,0	—	0,0
+ 15,00 +	9464,1	0,1917	0,2027	1,0293	93	9492,4	—	0,0
16,00	12451,0	0,1916	0,2027	1,0293	101	12373,6	—	0,0

	POWER DELIVERY							
SPEED [kt]	RPMPROP [RPM]	QPROP [kN·m]	QENG [kN·m]	PDPROP [kW]	PSPROP [kW]	PSTOTAL [kW]	PBTOTAL [kW]	TRANSP
5,00 !	31	106,20	106,20	335,0	345,4	690,7	690,7	—
7,00	43	201,74	201,74	883,5	910,8	1821,6	1821,6	—
9,00	55	325,81	325,81	1823,1	1879,5	3758,9	3758,9	—
10,00	61	398,98	398,98	2475,3	2551,8	5103,6	5103,6	984,3
11,00	67	480,86	480,86	3278,4	3379,8	6759,5	6759,5	817,5
12,00	73	573,79	573,79	4271,2	4403,3	8806,6	8806,6	684,5
13,00	80	681,74	681,74	5516,9	5687,5	11375,1	11375,1	574,1
14,00	86	810,57	810,57	7113,9	7334,0	14667,9	14667,9	479,5
+ 15,00 +	93	968,16	968,16	9207,7	9492,4	18984,9	18984,9	396,9
16,00	101	1164,29	1164,29	12002,4	12373,6	24747,2	24747,2	324,8

	EFFICIENCY				THRUST	
SPEED [kt]	EFFO	EFFG	EFFOA	MERIT	THRPROP [kN]	DELTHR [kN]
5,00 !	0,5055	1,0000	0,4993	0,29308	84,09	134,09
7,00	0,5021	1,0000	0,4955	0,28398	157,17	250,62
9,00	0,4993	1,0000	0,4923	0,27718	250,67	399,71
10,00	0,4982	1,0000	0,4912	0,27479	305,58	487,28
11,00	0,4977	1,0000	0,4905	0,27357	367,45	585,93
12,00	0,4980	1,0000	0,4907	0,27418	438,97	699,99
13,00	0,4994	1,0000	0,4920	0,27745	524,77	836,80
14,00	0,5021	1,0000	0,4946	0,28419	631,72	1007,34
+ 15,00 +	0,5062	1,0000	0,4985	0,29496	769,12	1226,43
16,00	0,5109	1,0000	0,5031	0,30995	948,62	1512,67

Report ID20170603-1208 HydroComp NavCad 2014 14.02.0029.S1002.539

Figura 4-1 – Datos Potencia hélice buque base

Fuente: Software NavCAD

Propulsion

3 Jun 2017 12:10
HydroComp NavCad 2014

Project ID	Petrolero 80.000 TPM
Description	Cuaderno 1
File name	Cuaderno 1.hcnc

Analysis parameters

Hull-propulsor interaction		System analysis	
Technique:	[Calc] Prediction	Cavitation criteria:	Keller eqn
Prediction:	Holtrop	Analysis type:	Free run
Reference ship:		CPP method:	
Max prop diam:	8500,0 mm	Engine RPM:	
Corrections		Mass multiplier:	
Viscous scale corr:	[On] Custom	RPM constraint:	
Rudder location:	Behind propeller	Limit [RPM/s]:	
Friction line:	ITTC-57	**Water properties**	
Hull form factor:	1,327	Water type:	Salt
Corr allowance:	0,000163	Density:	1026,00 kg/m3
Roughness [mm]:	[Off] 0,00	Viscosity:	1,18920e-6 m2/s
Ducted prop corr:	[Off]		
Tunnel stern corr:	[Off]		
Effective diam:			
Recess depth:			

Predicted propulsion

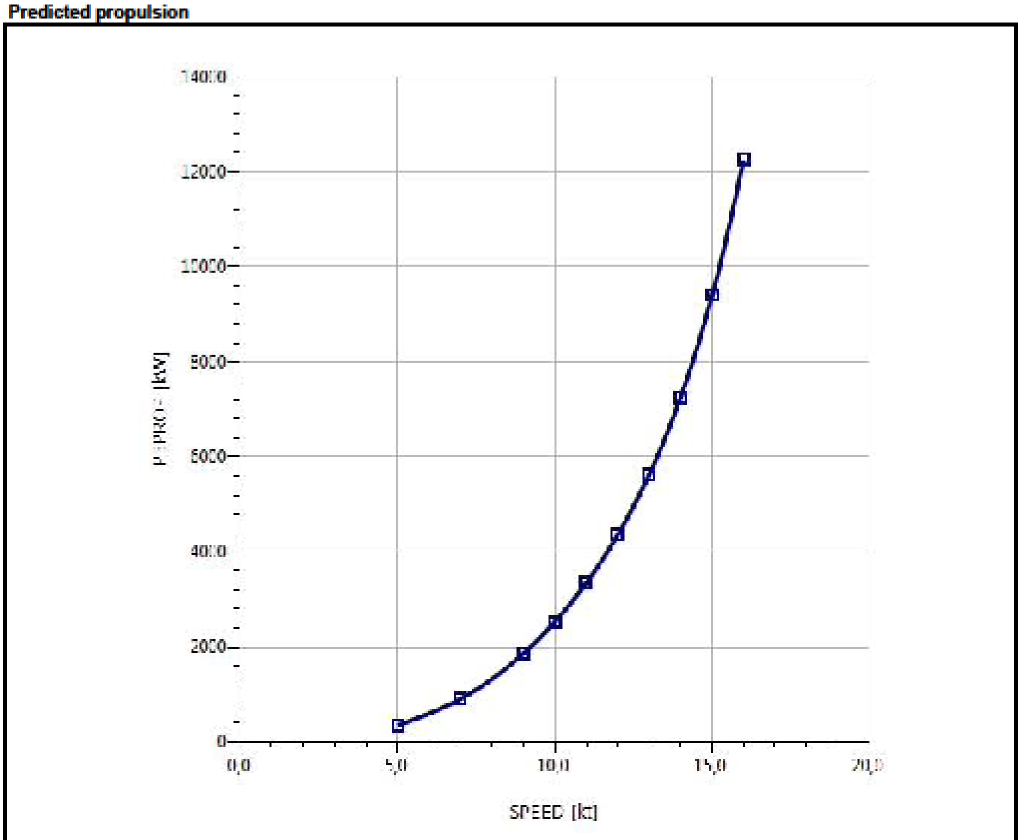


Figura 4-2 – Gráfica Potencia hélice buque base

Fuente: Software NavCAD

Vemos que para la velocidad de diseño (15 nudos) nos solicita una demanda total de 18.984,9 kW. Es decir, unas BHP (potencia al freno) de 9.492,4 kW

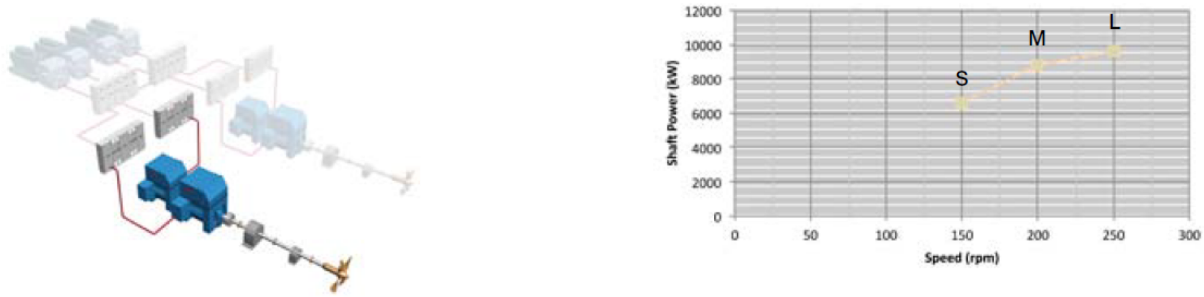
4.2. Elección preliminar del motor eléctrico buque base

Con la potencia BHP resultante, escogemos un modelo de motor eléctrico de la casa ABB.

Al ser nuestro sistema propulsivo de dos líneas de ejes, dispondremos de dos motores eléctricos, uno para cada línea de eje.

$$Potencia\ por\ eje = 9.492{,}4\ kW$$

Por tanto cada motor eléctrico deberá tener como potencia mínima 9.492,4 kW Entrando en la casa ABB, seleccionamos el motor eléctrico más conveniente.



Direct Drive 1000 - Tandem Drive					
Drive Step	S	M	L	X	Y
Propeller Speed (rpm)	≥150	≥200	≥250	-	-
Maximum Power (kW)	6600	8800	9600	-	-
Maximum Torque (kNm)	419.8	419.4	365.4	-	-

Figura 4-3 – Elección preliminar motor eléctrico

Fuente: ABB System Project Guide Passenger Vessels

Elegimos en una primera instancia el motor eléctrico de acople directo Direct Drive 1000 tandem drive de tamaño L a 250 rpm que nos proporciona una potencia de 9.600 kW cada uno, suficientes para la potencia que necesitamos en esta fase de diseño.

4.3. ESTIMACIÓN PRELIMINAR DEL DIÁMETRO DE HÉLICE

Una vez estimados los motores eléctricos, calculamos el diámetro de la hélice acorde a la potencia preliminar para poder iterar y aproximarnos al valor real que necesitamos.

Por lo tanto mediante el método de “By power” del software NavCAD, dimensionaremos la hélice mediante la función “Size”

Para ello las características proporcionadas al software han sido entre otras:

- Número de líneas de ejes: 2
- Acople directo motor hélice: No hay reductora, ratio igual a 1
- Diámetro de hélice: “Dimensionar”
- Punto de diseño de 1,03
- Potencia de referencia: 19.200 que nos daban los dos motores.
- Velocidad de diseño: 15 nudos

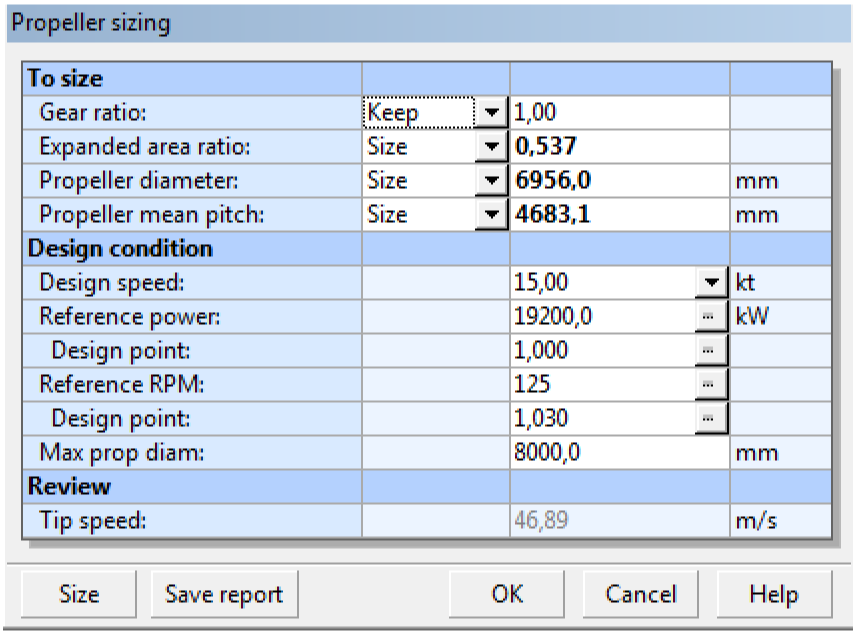


Figura 4-4 – Dimensionamiento hélice

Fuente: Software NavCAD

Como podemos observar, el software nos dimensiona el diámetro de la hélice con valores muy cercanos al cálculo preliminar que hicimos con anterioridad (Ver Capítulo 4):

Diámetro de hélice: 6956 mm, por lo tanto y como no existen hélices con ese diámetro de tipo comercial o sería demasiado costoso su fabricación, nos decantamos por unas hélices de 7 metros de diámetro.

Propulsion

3 jun 2017 01:21
HydroComp NavCad 2014

Project ID	Petrolero 80.000 TPM
Description	Cuaderno 1
File name	Cuaderno 1.hcnc

Analysis parameters

Hull-propulsor interaction	
Technique:	[Calc] Prediction
Prediction:	Holtrop
Reference ship:	
Max prop diam:	8000,0 mm
Corrections	
Viscous scale corr:	[On] Custom
Rudder location:	Behind propeller
Friction line:	ITTC-57
Hull form factor:	1,327
Corr allowance:	0,000163
Roughness [mm]:	[Off] 0,00
Ducted prop corr:	[Off]
Tunnel stern corr:	[Off]
Effective diam:	
Recess depth:	

System analysis	
Cavitation criteria:	Keller eqn
Analysis type:	Free run
CPP method:	
Engine RPM:	
Mass multiplier:	
RPM constraint:	
Limit [RPM/s]:	
Water properties	
Water type:	Salt
Density:	1026,00 kg/m3
Viscosity:	1,18920e-6 m2/s

Prediction method check [Holtrop]

Parameters	FN [design]	CP	LWL/BWL	BWL/T
Value	0,16	0,85	6,65	2,27
Range	0,06--0,80	0,55--0,85	3,90--14,90	2,10--4,00

Prediction results [System]

	HULL-PROPULSOR				ENGINE			
SPEED [kt]	PETOTAL [kW]	WFT	THD	EFFR	RPMENG [RPM]	PBPROP [kW]	FUEL [L/h]	LOADENG [%]
5,00 !	344,9	0,1992	0,2069	1,0204	35	300,2	—	3,1
7,00	902,5	0,1984	0,2069	1,0204	48	783,8	—	8,2
9,00	1850,7	0,1979	0,2069	1,0204	61	1604,9	—	16,7
10,00	2506,8	0,1977	0,2069	1,0204	68	2172,9	—	22,6
11,00	3315,7	0,1974	0,2069	1,0204	74	2873,8	—	29,9
12,00	4321,3	0,1973	0,2069	1,0204	81	3746,8	—	39,0
13,00	5596,4	0,1971	0,2069	1,0204	88	4858,1	—	50,6
14,00	7255,2	0,1970	0,2069	1,0204	96	6313,1	—	65,8
+ 15,00 +	9464,1	0,1968	0,2069	1,0204	104	8270,1	—	86,1
16,00	12451,0	0,1967	0,2069	1,0204	114	10956,1	—	114,1

	POWER DELIVERY							
SPEED [kt]	RPMPROP [RPM]	QPROP [kN·m]	QENG [kN·m]	PDPROP [kW]	PSPROP [kW]	PSTOTAL [kW]	PBTOTAL [kW]	TRANSP
5,00 !	35	80,33	80,33	285,4	291,2	582,5	600,5	—
7,00	48	151,46	151,46	745,1	760,3	1520,6	1567,6	—
9,00	61	243,20	243,20	1525,6	1556,7	3113,4	3209,7	—
10,00	68	297,20	297,20	2065,6	2107,7	4215,4	4345,8	—
11,00	74	357,82	357,82	2731,8	2787,6	5575,2	5747,6	961,4
12,00	81	427,20	427,20	3561,7	3634,4	7268,9	7493,7	804,4
13,00	88	509,01	509,01	4618,1	4712,3	9424,7	9716,2	672,1
14,00	96	608,67	608,67	6001,2	6123,7	12247,3	12626,1	557,0
+ 15,00 +	104	733,47	733,47	7861,6	8022,0	16044,0	16540,2	455,6
16,00	114	892,51	892,51	10414,9	10627,4	21254,8	21912,2	366,8

	EFFICIENCY				THRUST	
SPEED [kt]	EFFO	EFFG	EFFOA	MERIT	THRPROP [kN]	DELTHR [kN]
5,00 !	0,5978	0,9700	0,5921	0,42729	84,53	134,08
7,00	0,5998	0,9700	0,5935	0,41827	158,00	250,62
9,00	0,6012	0,9700	0,5944	0,41146	251,99	399,71
10,00	0,6016	0,9700	0,5947	0,40904	307,19	487,28
11,00	0,6018	0,9700	0,5947	0,4078	369,38	585,93
12,00	0,6017	0,9700	0,5945	0,40843	441,28	699,99
13,00	0,6011	0,9700	0,5938	0,41173	527,53	836,80
14,00	0,5998	0,9700	0,5924	0,41848	635,05	1007,35
+ 15,00 +	0,5973	0,9700	0,5899	0,42914	773,17	1226,44
16,00	0,5933	0,9700	0,5858	0,44371	953,61	1512,67

Report ID20170603-1321 HydroComp NavCad 2014 14.02.0029.S1002.539

Figura 4-5 – Datos dimensionamiento hélice

Fuente: Software NavCAD

Propulsion
3 Jun 2017 01:22
HydroComp NavCad 2014

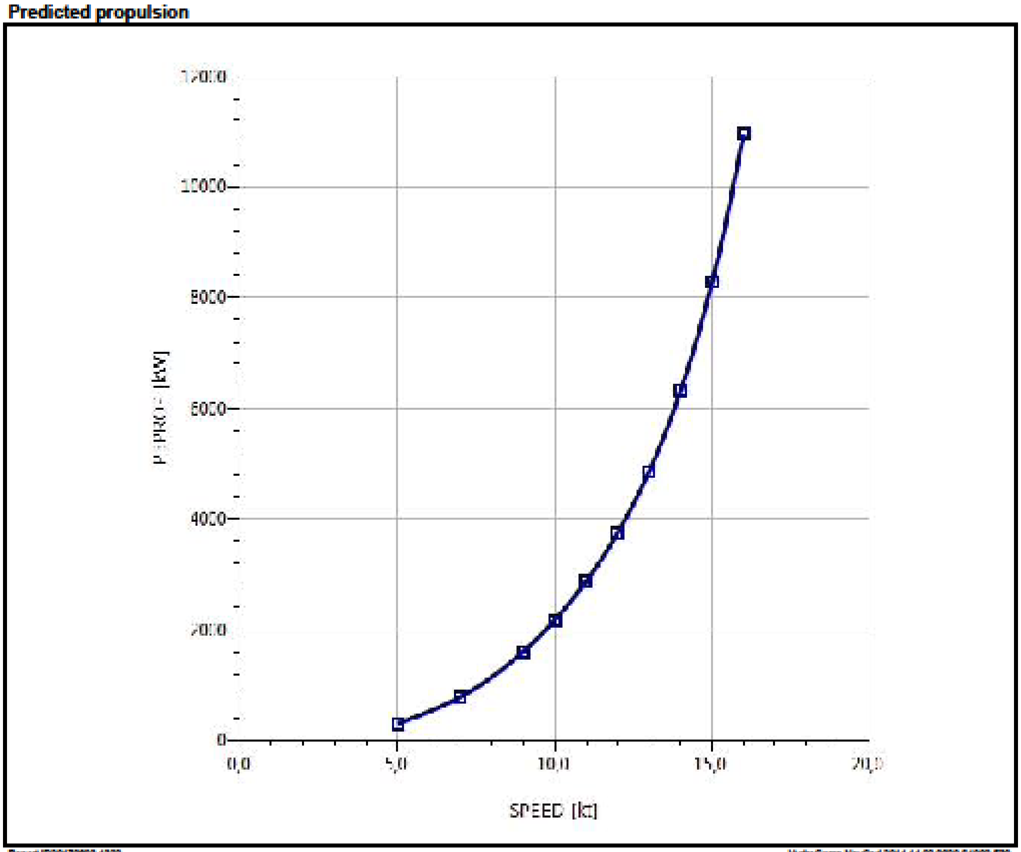
Project ID	Petrolero 80.000 TPM
Description	Cuaderno 1
File name	Cuaderno 1.hcnc

Analysis parameters

Hull-propulsor interaction		
Technique:	[Calc]	Prediction
Prediction:		Holtrop
Reference ship:		
Max prop diam:		8000,0 mm
Corrections		
Viscous scale corr:	[On]	Custom
Rudder location:		Behind propeller
Friction line:		ITTC-57
Hull form factor:		1,327
Corr allowance:		0,000163
Roughness [mm]:	[Off]	0,00
Ducted prop corr:	[Off]	
Tunnel stern corr:	[Off]	
Effective diam:		
Recess depth:		

System analysis	
Cavitation criteria:	Keller eqn
Analysis type:	Free run
CPP method:	
Engine RPM:	
Mass multiplier:	
RPM constraint:	
Limit [RPM/s]:	
Water properties	
Water type:	Salt
Density:	1026,00 kg/m3
Viscosity:	1,18920e-6 m2/s

Predicted propulsion



Report ID20170603-1322 HydroComp NavCad 2014 14.02.0029.S1002.539

Figura 4-6 – Gráfica dimensionamiento hélice

Fuente: Software NavCAD

Otro método seria mediante la entrada con los BHP de nuestro motor en la siguiente gráfica.

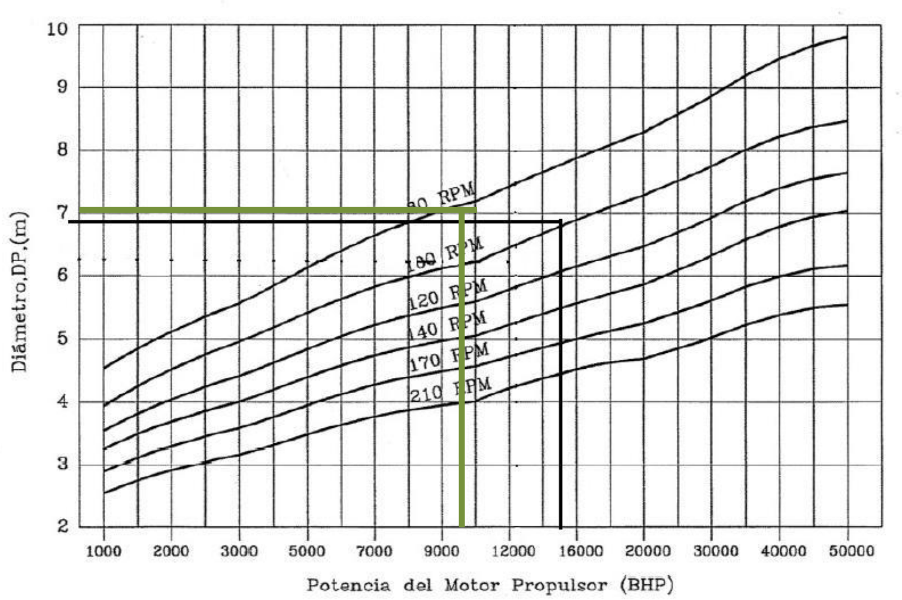


Figura 4-7 - grafica BHP-RPM

Fuente: El proyecto básico del buque mercante

Mediante la gráfica del libro “El proyecto básico del buque mercante”, nos indica que para la potencia por eje y diámetro de hélice, las rpm deberían ser cercanas a 80 rpm

4.4. PREDICCIÓN DE POTENCIA

Con el dato inicial de las BHP de la potencia de los motores eléctricos (9.600 kW cada uno), que nos da un diámetro de 7 metros, volvemos a iterar y calcular la predicción de potencia para afinar el resultado de la potencia ya con el diámetro de hélice definido.

Para ello las características proporcionadas al software han sido entre otras:

- Número de líneas de ejes: 2
- Numero de palas: 4
- Gear efficiency (Acople directo motor hélice): No hay reductora, ratio igual a 1
- Diámetro de hélice de 7 metros
- Shaft efficiency (Rendimiento del eje): 0,98 al ser múltiple
- Punto de diseño de 1,03

Engine/gear		
Engine data:	None defined	
Rated RPM:		RPM
Rated power:		kW
Gear efficiency:	1,000	
Load correction:	Off	
Gear ratio:	1,000	
Shaft efficiency:	0,980	

Propeller sizing

To size			
Gear ratio:	Keep	1,00	
Expanded area ratio:	Size	**0,300**	
Propeller diameter:	Keep	7000,0	mm
Propeller mean pitch:	Size	**4613,1**	mm

Figura 4-8 – Datos hélice

Fuente: Software NavCAD

Propulsion
3 jun 2017 01:55
HydroComp NavCad 2014

Project ID Petrolero 80.000 TPM
Description Cuaderno 1
File name Cuaderno 1.hcnc

Analysis parameters

Hull-propulsor interaction		System analysis	
Technique:	[Calc] Prediction	Cavitation criteria:	Keller eqn
Prediction:	Holtrop	Analysis type:	Free run
Reference ship:		CPP method:	
Max prop diam:	8000,0 mm	Engine RPM:	
Corrections		Mass multiplier:	
Viscous scale corr:	[On] Custom	RPM constraint:	
Rudder location:	Behind propeller	Limit [RPM/s]:	
Friction line:	ITTC-57	**Water properties**	
Hull form factor:	1,327	Water type:	Salt
Corr allowance:	0,000163	Density:	1026,00 kg/m3
Roughness [mm]:	[Off] 0,00	Viscosity:	1,18920e-6 m2/s
Ducted prop corr:	[Off]		
Tunnel stern corr:	[Off]		
Effective diam:			
Recess depth:			

Prediction method check [Holtrop]

Parameters	FN [design]	CP	LWL/BWL	BWL/T
Value	0,16	0,85	6,65	2,27
Range	0,06--0,80	0,55--0,85	3,90--14,90	2,10--4,00

Prediction results [System]

	HULL-PROPULSOR				ENGINE			
SPEED [kt]	PETOTAL [kW]	WFT	THD	EFFR	RPMENG [RPM]	PBPROP [kW]	FUEL [L/h]	LOADENG [%]
5,00 !	344,9	0,1992	0,2069	1,0209	34	307,3	—	3,2
7,00	902,5	0,1984	0,2069	1,0209	47	803,5	—	8,4
9,00	1850,7	0,1979	0,2069	1,0209	60	1647,2	—	17,2
10,00	2506,8	0,1977	0,2069	1,0209	67	2231,2	—	23,2
11,00	3315,7	0,1974	0,2069	1,0209	73	2951,6	—	30,7
12,00	4321,3	0,1973	0,2069	1,0209	80	3847,8	—	40,1
13,00	5596,4	0,1971	0,2069	1,0209	87	4986,2	—	51,9
14,00	7255,2	0,1970	0,2069	1,0209	94	6471,8	—	67,4
+ 15,00 +	9464,1	0,1968	0,2069	1,0209	103	8462,2	—	88,1
16,00	12451,0	0,1967	0,2069	1,0209	112	11181,7	—	116,5

	POWER DELIVERY						
SPEED [kt]	RPMPROP [RPM]	QPROP [kN·m]	QENG [kN·m]	PDPROP [kW]	PSPROP [kW]	PSTOTAL [kW]	PBTOTAL [kW]	TRANSP
5,00 !	34	83,66	83,66	292,1	298,1	596,2	614,7	—
7,00	47	158,08	158,08	763,9	779,4	1558,9	1607,1	—
9,00	60	254,26	254,26	1565,9	1597,8	3195,6	3294,5	—
10,00	67	310,90	310,90	2121,0	2164,3	4328,6	4462,5	—
11,00	73	374,43	374,43	2805,8	2863,0	5726,1	5903,2	936,1
12,00	80	446,96	446,96	3657,7	3732,4	7464,8	7695,6	783,3
13,00	87	532,12	532,12	4739,8	4836,6	9673,1	9972,3	654,9
14,00	94	635,24	635,24	6152,1	6277,6	12555,3	12943,6	543,3
+ 15,00 +	103	763,48	763,48	8044,2	8208,3	16416,7	16924,4	445,2
16,00	112	925,65	925,65	10629,3	10846,3	21692,5	22363,4	359,4

	EFFICIENCY				THRUST	
SPEED [kt]	EFFO	EFFG	EFFOA	MERIT	THRPROP [kN]	DELTHR [kN]
5,00 !	0,5838	0,9700	0,5785	0,41468	84,53	134,08
7,00	0,5848	0,9700	0,5789	0,40528	157,99	250,62
9,00	0,5854	0,9700	0,5791	0,39821	251,98	399,71
10,00	0,5856	0,9700	0,5791	0,39571	307,19	487,28
11,00	0,5857	0,9700	0,5791	0,39442	369,38	585,93
12,00	0,5856	0,9700	0,5789	0,39506	441,28	699,99
13,00	0,5854	0,9700	0,5785	0,39849	527,54	836,80
14,00	0,5848	0,9700	0,5779	0,4055	635,05	1007,35
+ 15,00 +	0,5835	0,9700	0,5765	0,41661	773,17	1226,44
16,00	0,5811	0,9700	0,5740	0,43187	953,61	1512,67

Report ID20170603-1355 HydroComp NavCad 2014 14.02.0029.S1002.539

Figura 4-9 – Datos BHP hélice definida

Fuente: Software NavCAD

Propulsion

3 Jun 2017 02:04
HydroComp NavCad 2014

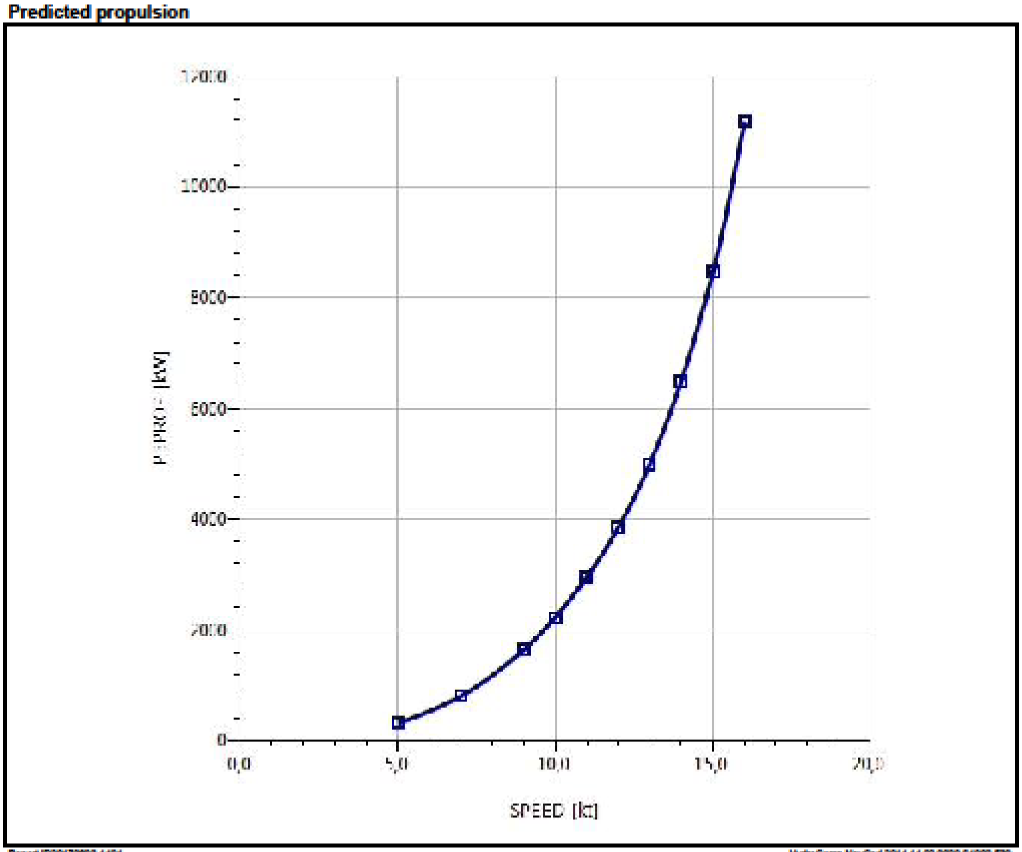
Project ID	Petrolero 80.000 TPM
Description	Cuaderno 1
File name	Cuaderno 1.hcnc

Analysis parameters

Hull-propulsor interaction	
Technique:	[Calc] Prediction
Prediction:	Holtrop
Reference ship:	
Max prop diam:	8000,0 mm
Corrections	
Viscous scale corr:	[On] Custom
Rudder location:	Behind propeller
Friction line:	ITTC-57
Hull form factor:	1,327
Corr allowance:	0,000163
Roughness [mm]:	[Off] 0,00
Ducted prop corr:	[Off]
Tunnel stern corr:	[Off]
Effective diam:	
Recess depth:	

System analysis	
Cavitation criteria:	Keller eqn
Analysis type:	Free run
CPP method:	
Engine RPM:	
Mass multiplier:	
RPM constraint:	
Limit [RPM/s]:	
Water properties	
Water type:	Salt
Density:	1026,00 kg/m3
Viscosity:	1,18920e-6 m2/s

Predicted propulsion



Report ID20170603-1404
HydroComp NavCad 2014 14.02.0029.S1002.539

Figura 4-10 – Gráfica BHP hélice definida

Fuente: Software NavCAD

Vemos que para la velocidad de diseño (15 nudos) para nuestra hélice definida, nos solicita una demanda total de 16.924,4 kW. Es decir, unas BHP (potencia al freno) de 8.462,2 kW por cada eje.

Si comparamos dicha potencia con la estimación empírica al principio de este cuaderno/documento, vemos que es ligeramente superior (16.304,04 kW)

Necesitaremos por tanto dos motores eléctricos que su potencia mínima sea de 8.462,2 kW, sin embargo no es recomendable que el motor eléctrico vaya al 100% de su carga/capacidad por lo que le daremos un margen de funcionamiento entre un 90 y un 95% de carga, por lo tanto:

$$Potencia\ mín = 16.924,4\ \ kW$$

$$Potencia\ mín = \frac{16.924,4}{\eta * n}\ \ kW$$

Siendo:

- n: número de motores
- η : rendimiento (90-95%)

$$Potencia\ 90\% = \frac{16.924,4}{0,90 * 2} =\ 9.402,33\ kW\ por\ eje$$

$$Potencia\ 95\% = \frac{16.924,4}{0,95 * 2} =\ 8.907,57\ kW\ por\ eje$$

Por lo tanto tendremos que buscar un motor dentro de ese margen de funcionamiento.

Capítulo 5. MOTOR ELÉCTRICO

Una vez definidos la potencia y el diámetro de la hélice que necesitaremos en el buque, tenemos que definir los motores eléctricos que proporcionaran, en energía mecánica para el accionamiento de las hélices.

Para ello hemos seguido la guía del sistema de propulsión diesel eléctrica de la más que famosa compañía ABB.

En primer lugar, definiremos el tipo de sistema de propulsión que tendrá nuestro buque, en nuestro caso constara de 4 diesel generadores conectados a 2 motores eléctricos.

A continuación se puede apreciar el esquema del sistema a utilizar:

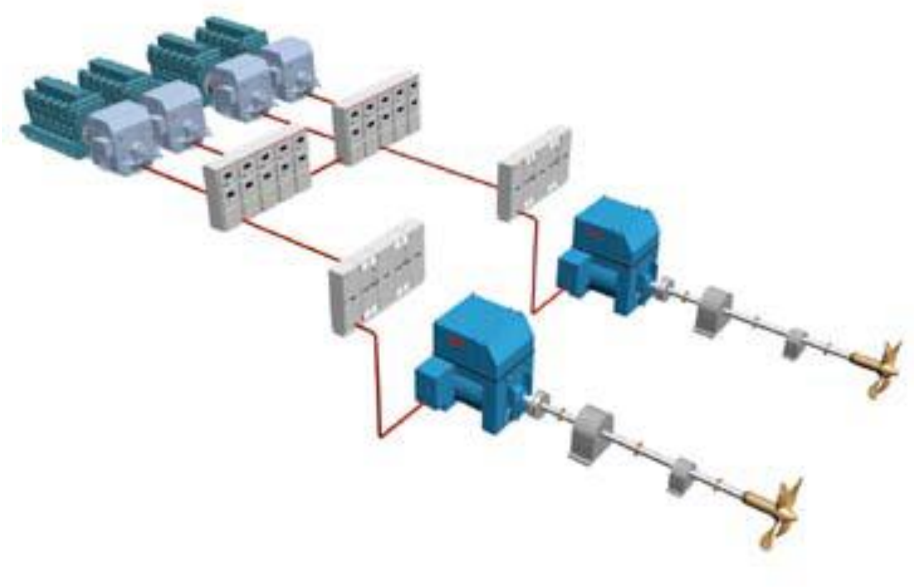


Figura 5-1 - Esquema sistema Direct drive de ABB

Fuente: ABB System Project Guide Passenger Vessels

Para la selección del motor eléctrico seguiremos los pasos que nos indica ABB en su catálogo: ABB System Project Guide Passenger Vessels

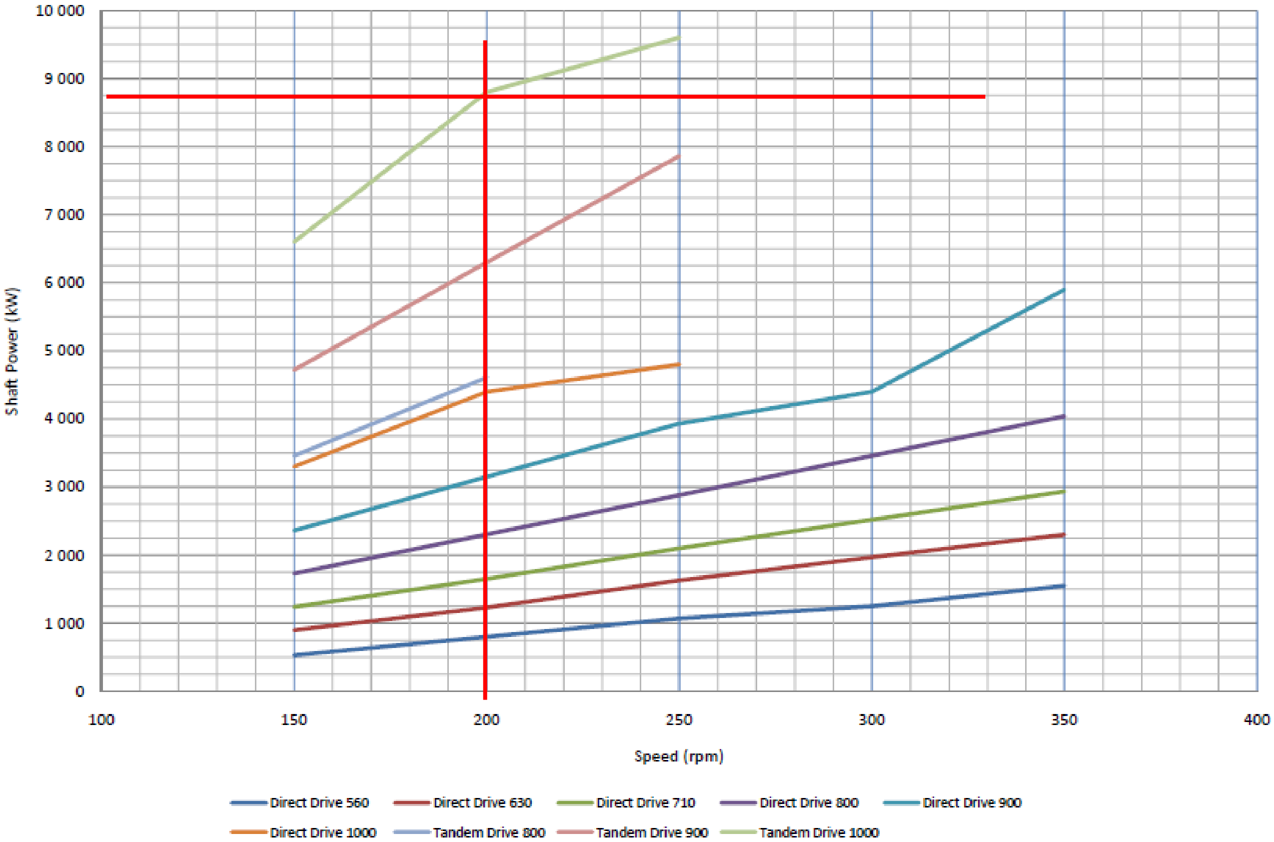
1. En primer lugar, seleccionaremos el tipo de plataforma que será el sistema de propulsión. En nuestro caso, hemos seleccionado el sistema Direct drive debido a que es el más eficiente de los disponibles

Figura 5-2 – Plataforma ABB

Fuente: ABB System Project Guide Passenger Vessels

Hemos elegido el Direct drive al ser el más sencillo, más eficiente y el que más rango de potencias ofrece.

2. En Segundo lugar, seleccionamos el modelo del motor eléctrico para nuestra potencia de 8.907,57 kW



Performance limits for Direct Drives 1120 – 2000. (Figure 2)

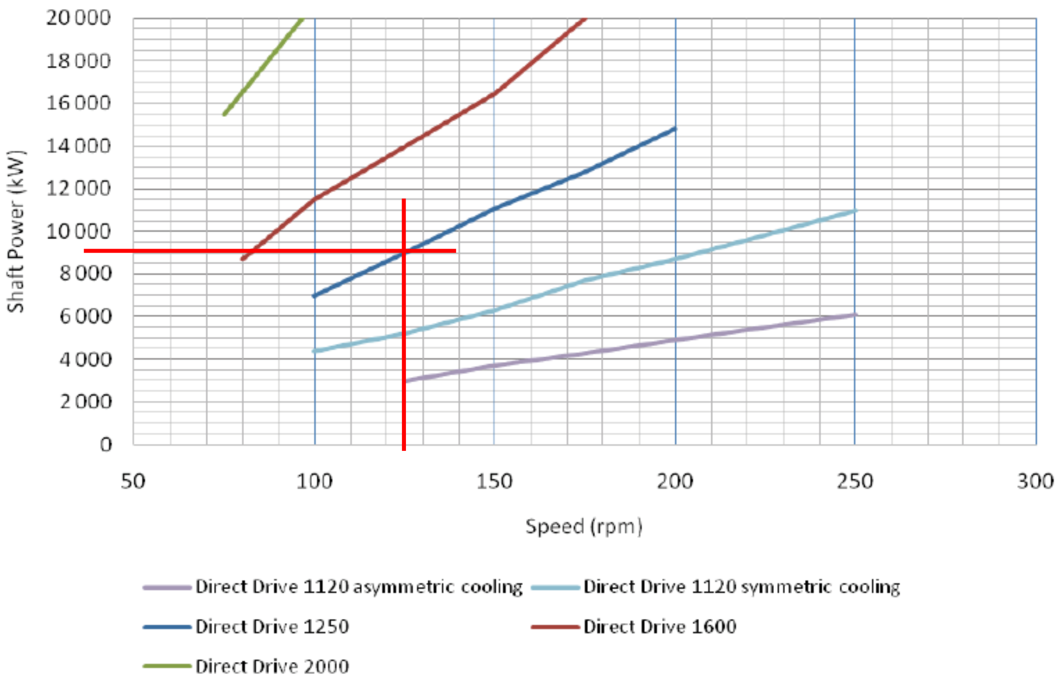


Figura 5-3 – Rango de potencia ABB

Fuente: ABB System Project Guide Passenger Vessels

Para nuestra potencia de 16.924,4 kW que necesitamos, necesitaremos por tanto 2 motores eléctricos de 8.907,57 kW mínimo cada uno.

Vemos que tenemos las posibilidades de elegir dentro del rango de potencias de nuestro sistema:

- Tamdem drive 100
- Direct Drive 1250

3. Selección del sistema. Elegimos el sistema Single Drive por ser el más eficiente

Figura 5-4 – Sistema de potencia ABB

Fuente: ABB System Project Guide Passenger Vessels

<u>Componentes del sistema Single Drive</u>

Number of main components – Direct Drive

	Single Drive	Single Drive (AFE)	Single Drive (with Trans-former)	Tandem Drive	Single Drive (twin in/out)	Full Redundant Drive
Motor	1	1	1	2	1	1
Frequency converter	1	1	1	2	1	2
Braking resistor	1	0	1	2	1	2
Transformer	0	0	1	0	2	2
Excitation transformer	0, 1)	0	0, 1)	0	1	2
Harmonic filter	1	0	0	1	0	0

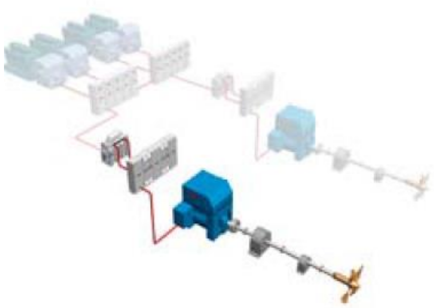
1) One excitation transformer in case of a synchronous motor

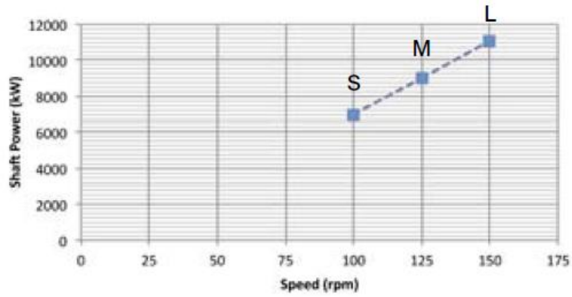
Figura 5-5 – Componentes del sistema Single Drive de ABB

Fuente: ABB System Project Guide Passenger Vessels

4. Selección del modelo del motor:

De entre los 2 posibles sistemas, hemos elegido el Direct Drive 1150 debido a que es el que nos ofrece la potencia máxima más cercana a la que necesitamos a menor número de vueltas (rpm)





Direct Drive 1250 - Single Drive (with Transformer)				
	Drive Step	S	M	L
	Propeller Speed (rpm)	≥100	≥125	≥150
	Maximum Power (kW)	7000	9000	11100
	Maximum Torque (kNm)	669,0	687,5	706,6
	Drive (kVA)	9000	11000	14000
	Transformer (kVA)	8000	12000	14000
	Braking Capacity (MJ)	30	46	46
Drivetrain Efficiency (%)	Motor	94,69	95,6	96,53
	Frequency Converter	98,5	98,5	98,5
	Transformer	99	99	99
	Total Electrical Efficiency	92,3	93,2	94,1
Main Connection	Input Voltage (VAC)	6600 / 11000	6600 / 11000	6600 / 11000
	Frequency (Hz)	50/60	50/60	50/60
	Power factor	0,95	0,95	0,95
	Input power (kVA)	7980	10162	12413
	Input Current (A)	699 / 419	889 / 534	1086 / 652
Footprint (m²)	Motor	24,4	24,4	24,4
	Frequency Converter	9,3	9,8	12,7
	Braking Resistor	1,6	2,0	2,0
	Transformer	10,9	12,5	13,3
	Excitation Transformer	0,8	0,8	1,5
	Harmonic Filter	-	-	-
	Total	47,0	49,5	53,8
Dimensions (L x W xH)	Motor	5940 x 4100 x 3925	5940 x 4100 x 3925	5940 x 4100 x 3925
	Frequency Converter	7930 x 1176 x 2475	8330 x 1176 x 2475	10630 x 1176 x 2475
	Braking Resistor	1800 x 900 x 1700	2200 x 900 x 1700	2200 x 900 x 1700
	Transformer	4200 x 2600 x 2950	5000 x 2500 x 3300	5300 x 2500 x 3500
	Excitation Transformer	1230 x 670 x 1355	1230 x 670 x 1355	1240 x 1170 x 1555
	Harmonic Filter	-	-	-
Weight (kg)	Motor	84050	84100	84100
	Frequency Converter	6600	6800	8900
	Braking Resistor	620	750	750
	Transformer	12500	17200	18500
	Excitation Transformer	1330	1330	1720
	Harmonic Filter	-	-	-
	Total	103770	106850	112250

Figura 5-6 – Características Motor ABB

Fuente: ABB System Project Guide Passenger Vessels

En resumen los dos motores eléctricos elegidos y que proporcionaran la energía mecánica suficiente a cada una de las hélices son los motores cuyo modelo es **Direct Drive 1150 con transformador de paso M de 9.000 kW de potencia máxima a 125 rpm.**

Como datos a destacar se encuentran:

Descripción	Peso (kg)
Motor	84.100
Convertidor de frecuencia	6.800

Descripción	Peso (kg)
Resistor	750
Transformador	17.200
Trans. Excitación	1.330
Filtro harmonic	-
Total	**108.850**

Tabla 5-1– Pesos sistema Direct drive 1250M

Fuente: Propia a partir de ABB

Al ser dos líneas de ejes, el peso total será el doble:

$$P_{Sist.1250M} = 2 * 108.850 = 217{,}7\ toneladas$$

Capítulo 6. DIESEL GENERADORES

Aunque no es objeto propio de este documento, vamos a realizar una estimación de los diesel generadores necesarios para proporcionar y suministrar energía eléctrica a los motores, debido en gran parte, a la necesidad de conocer el peso de estos equipos

Una planta de Propulsión de diesel eléctrica está compuesta por los siguientes equipos, con sus rendimientos correspondientes, se han estimado los rendimientos medios de los motores eléctricos ABB y los rendimientos medios aproximados de los diesel generadores:

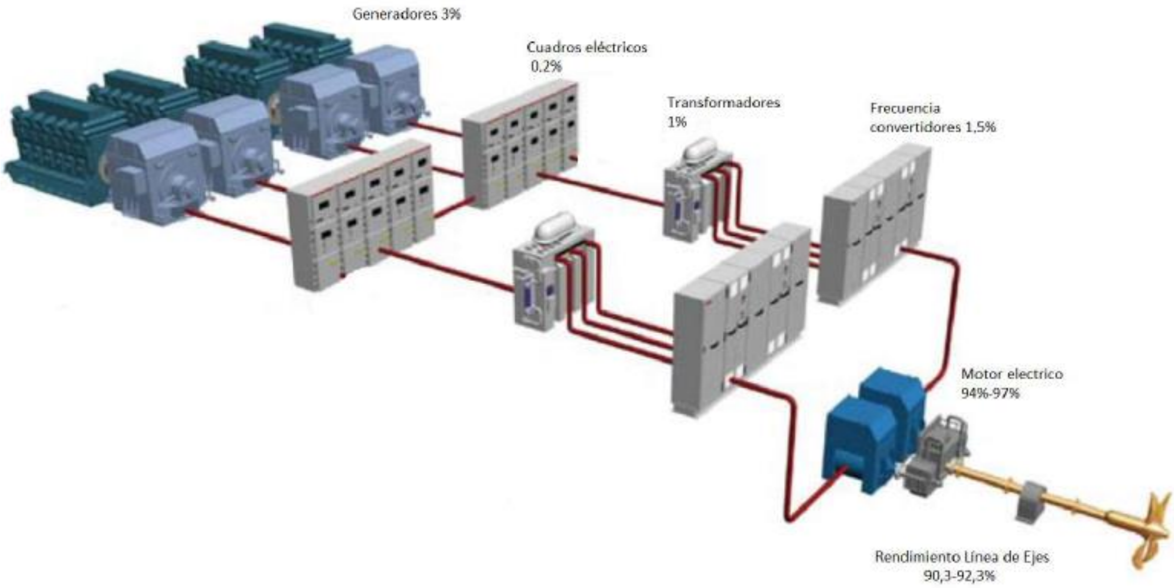


Figura 6-1 – Rendimientos prop. Diésel eléctrica

Fuente: ABB System Project Guide Passenger Vessels

Para asignar los márgenes de rendimientos anteriores a cada equipo se tomó como referencia

“Diesel-electric Propulsión Plants”.

Como hemos visto en el capítulo anterior, en esta planta se utilizarán 2 motores eléctricos acoplados al engranaje reductor estos dos motores eléctricos tendrán que estar dimensionados para soportar una carga eléctrica de 8.907,57 kW

Los motores eléctricos elegidos para este proyecto son de ABB ya que son motores de alta fiabilidad y nos permiten un menor mantenimiento e impacto ambiental.

Nuestro motor en cuestión es el Direct Drive 1150 con transformador de paso M de 9.000 kW de potencia máxima a 125 rpm.

6.1. CÁLCULO DIESEL GENERADORES

Aplicando los rendimientos antes mencionados, calcularemos la potencia necesaria que deberán proporcionar los diesel generadores.

- Rendimiento convertidores de frecuencia: 98,5%
- Rendimiento transformadores: 99%
- Rendimiento cuadros eléctricos: 99,8%

$$Potencia = \frac{9.000 * 2}{0,99 * 0,998 * 0,985} = 18.495,69\ kW$$

Teniendo en cuenta que el rendimiento de los diesel generadores es del 97%, la potencia será de:

$$Potencia = \frac{18495,69}{0,97} = 19.067,72\ kW$$

<u>Potencia eléctrica</u>

En el capítulo 10, definiremos con detalle el sistema de propulsión y energía del buque, por lo que para hacernos una idea de la potencia y pesos de los diesel generadores, estimaremos el resto de consumidores con la potencia eléctrica del buque base:

Potencia: 3 * 700= 2.100 kW

La potencia suministrada por los diesel generadores será la suma de la potencia necesaria para la propulsión y la potencia para satisfacer la demanda de los consumidores más la aplicación del porcentaje de funcionamiento óptimo de los diesel generadores, de MCR al 85%.

$$Pot_{generadores} = 21.167,72\ kW$$

$$Pot_{generadores} = \frac{21.167,72}{0,85} = 23.519,688\ kW$$

Los diesel generados instalados en esta cámara de máquinas tienen que tener la característica de ser diesel dual fuel, es decir, poder consumir HFO, MDO y gas/LNG, para cumplir con esas características, se elegirá un motor wartsila por tener una alta flexibilidad para cumplir esta característica.

Se instalarán cuatro diesel generadores y se cumplirá la normativa SOLAS que especifica:

“La capacidad de los grupos electrógenos debe ser tal que aunque uno cualquiera de ellos se pare, o deje de estar en funcionamiento, sea posible alimentar los servicios necesarios e indispensables para lograr condiciones operacionales normales de propulsión y seguridad”

SOLAS indica que la potencia y capacidad de los DDGG deberá ser suficiente como para que al disponer de 4 DDGG en nuestro caso, con solo tres de ellos, proporcionen los servicios mínimos de propulsión y funcionamiento de servicios vitales del buque, tales como propulsión, contraincendios, maniobrabilidad y rumbo, generación de agua… y por un tiempo suficiente como para permitir al buque desviarse de su rumbo y acudir al puerto más cercano o previsto para realizar una parada por mantenimiento.

A la hora de definir los DDGG, valoraremos que sean de la misma marca y modelo, ya que optimizaría costes en el mantenimiento/repuestos y nos darían ventajas como las siguientes mencionadas:

- Optimización en tiempo de instalación (mismos polines, maniobras, documentos de instalación…)
- Mismas características (mayor disponibilidad de repuestos)
- Menor tiempo de entrega

La potencia mínima de cada DDGG será de:

$$Pot._{generador} = \frac{23.519,688}{4} = 5879,922\ kW$$

Los motores duales serán de Wartsila, por lo que nos vamos a la página Web de la empresa y elegimos el modelo que cumpla las características.

Por estas razones, me he decidido a elegir cuatro diesel generadores 12V34DF. Estos diesel generadores, destacan por la capacidad para consumir desde HFO, a LFO e incluso gas LNG, por lo tanto no solo cumple con la potencia requerida, sino que también nos proporciona multitud de opciones para el combustible.

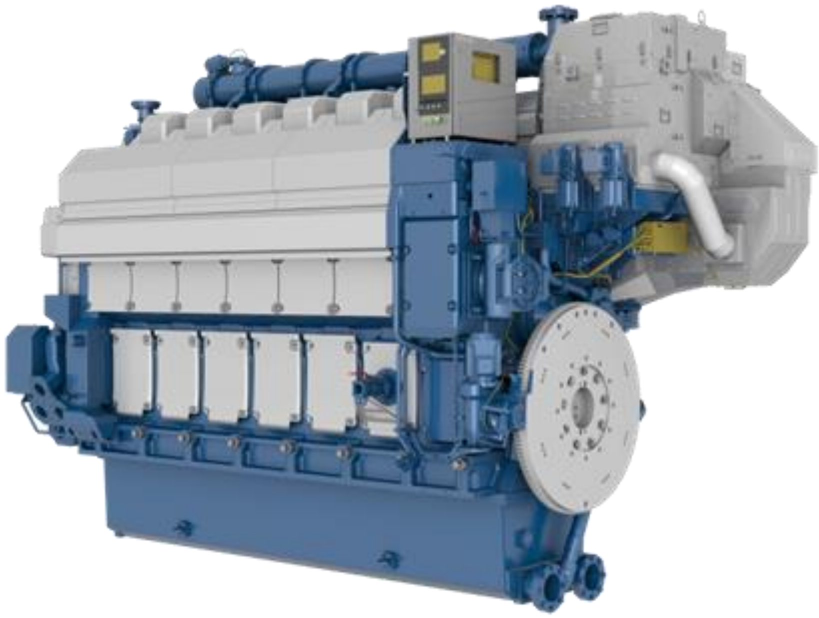


Figura 6-2 - Diesel generador dual fuel

Fuente: http://www.wartsila.com/products/marine-oil-gas/engines-generating-sets/dual-fuel-engines/wartsila-34df

Rated power	
Engine type	kW
6L34DF	3 000
8L34DF	4 000
9L34DF	4 500
12V34DF	6 000
16V34DF	8 000

Figura 6-3 - Potencia del diesel generador

Fuente: Propia

Los diesel generadores escogidos, cumplen uno de los requisitos de la RPA, que la propulsión sea duel fuel - diesel eléctrico. Los diesel generadores DFDE son motores que pueden funcionar con LNG y con diferentes combustibles: fuel oil pesado, diésel marino, gas e incluso, biodiesel.

Este tipo de propulsión es usualmente elegida por los armadores, debido a que son equipos fáciles de instalar con respecto a otras plantas.

Suelen instalarse para operar en modo gas o diesel. El tipo de combustible de funcionamiento se puede cambiar mientras el motor está en marcha, dentro de ciertos límites, sin interrumpir la generación de energía. Si el suministro de gas fallara, el motor pasaría automáticamente a la generación de energía mediante la combustión de diesel.

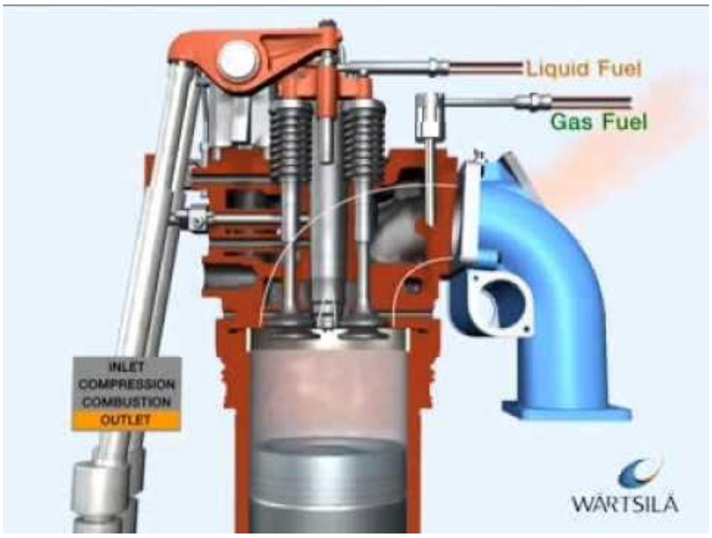


Figura 6-4 - Inyección motor DFDE

Fuente: https://www.youtube.com/watch?v=6mifHJ3MkfE

Como se puede ver en la imagen anterior el motor funciona con inyección de diesel o con inyección de gas de forma alterna, los gases expulsados son menos contaminantes los del LNG.
Los motores diesel duales pueden ser de 2T o de 4T, en función de las necesidades del buque. Si tomamos como ejemplo el suministrador Wartsilla podemos ver en su catálogo de productos los diferentes motores de 2T y 4T.

Toda la información sobre el motor puede verse en los anexos del presente documento, o bien en el enlace del suministrador indicado aquí: http://www.wartsila.com/products/marine-oil-gas/engines-generating-sets/diesel-engines/wartsila-34df

Nuestra planta de propulsión contará con cuatro motores duales (Fuel-Gas) del modelo 34/DF. Este tipo de motores tienen una amplia flexibilidad para alimentarse por HFO y LNG. Pudiendo así adaptarnos a las condiciones y cantidades de Boil-Off.
Otra ventaja de usar este tipo de motores es que se puede usar HFO MDO por lo que se pueden adaptar los motores a los precios de los combustibles en el mercado.

$$Wartsilla\ 12V34DF\ Potencia\ por\ cilindro = \frac{5.600}{12} = 466{,}666\ kW/cyl$$

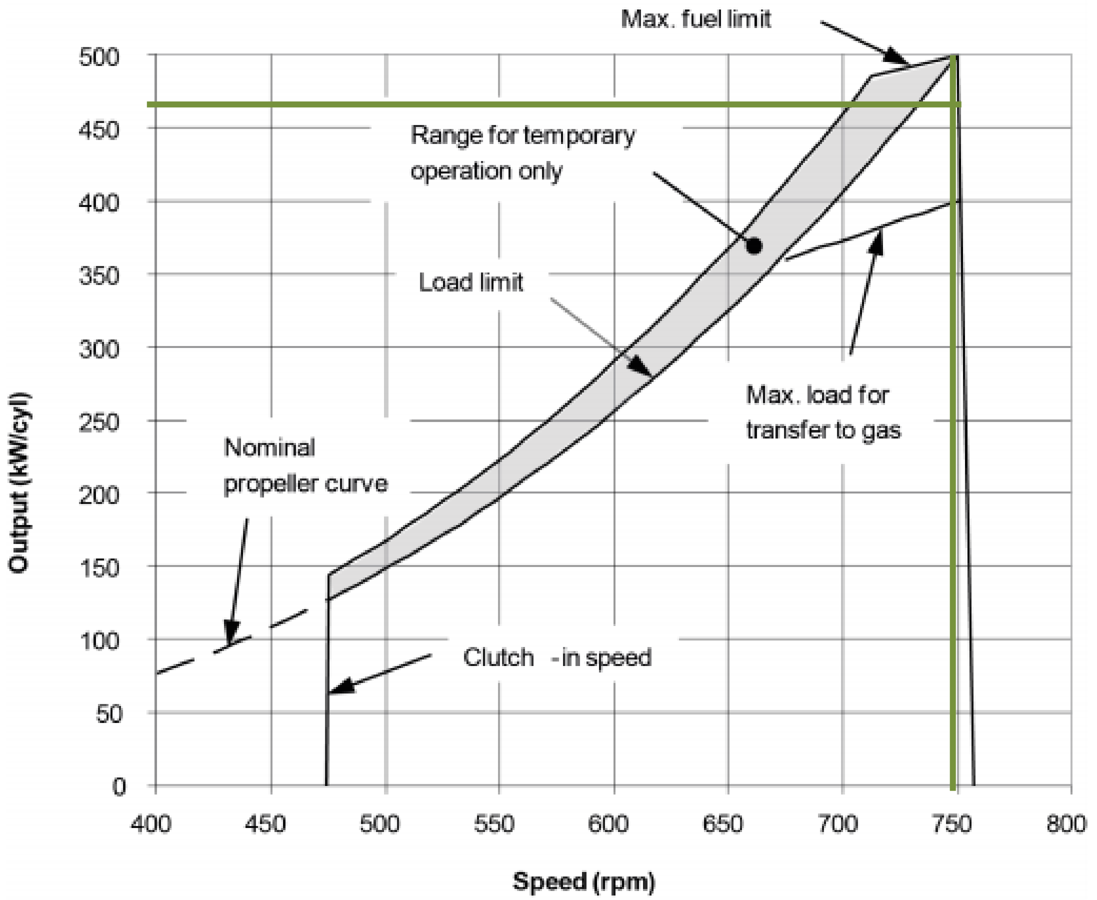


Figura 6-5 - Rango operación Wartsilla 34DF

Fuente: http://cdn.wartsila.com/docs/default-source/product-files/engines/df-engine/product-guide-o-e-w34df.pdf?sfvrsn=6

Wärtsilä 12V34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Engine speed	rpm	720		750		720		750		750		750	
Engine output	kW	5760		6000		5760		6000		6000		6000	
Mean effective pressure	MPa	2.2		2.2		2.2		2.2		2.2		2.2	
Speed mode		Constant		Constant		Constant		Constant		Constant		Variable	
IMO compliance		Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2
Combustion air system (Note 1)													
Flow at 100% load	kg/s	8.9	10.7	8.9	10.7	8.9	10.7	8.9	10.7	8.9	10.7	8.9	11.0
Temperature at turbocharger in-take, max.	°C	45		45		45		45		45		45	
Temperature after air cooler (TE 601), load > 70%	°C	45	-	45	-	45	-	45	-	45	-	45	-
Temperature after air cooler (TE 601), load 30...70%	°C	55	-	55	-	55	-	55	-	55	-	55	-
Temperature after air cooler (TE 601)	°C	-	50	-	50	-	50	-	50	-	50	-	50

Figura 6-6 - Características Wartsilla 12V34DF

Fuente: http://cdn.wartsila.com/docs/default-source/product-files/engines/df-engine/product-guide-o-e-w34df.pdf?sfvrsn=6

Dimensiones motor 12V34/DF

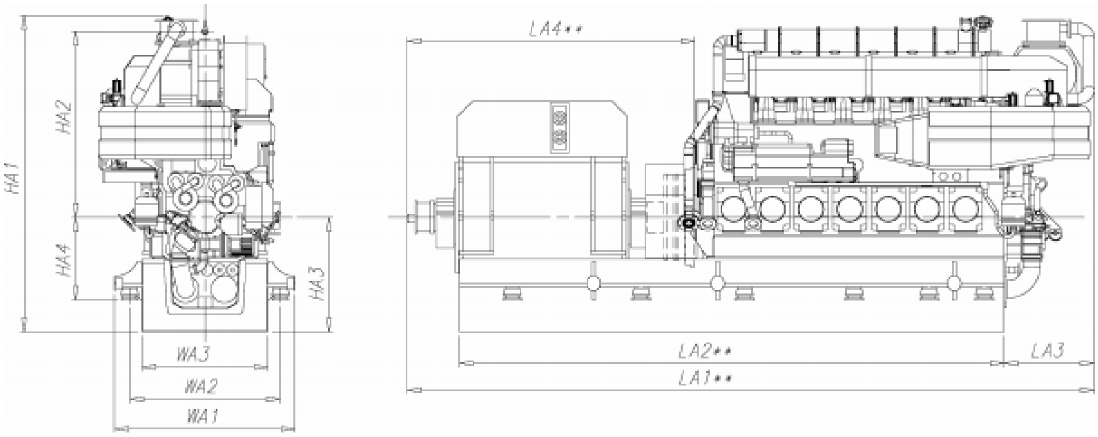


Fig 1-5 **In-line engines (DAAE082427)**

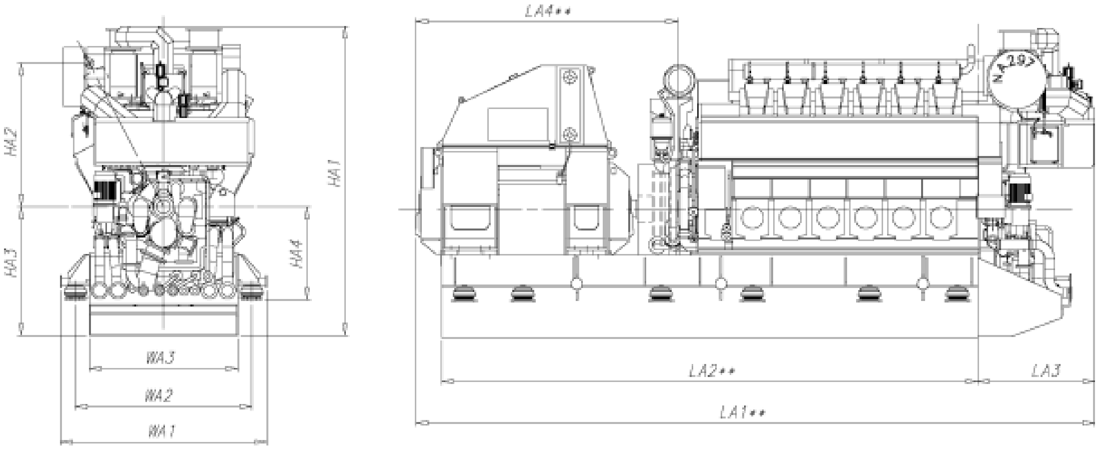


Fig 1-6 **V engines (DAAE082975)**

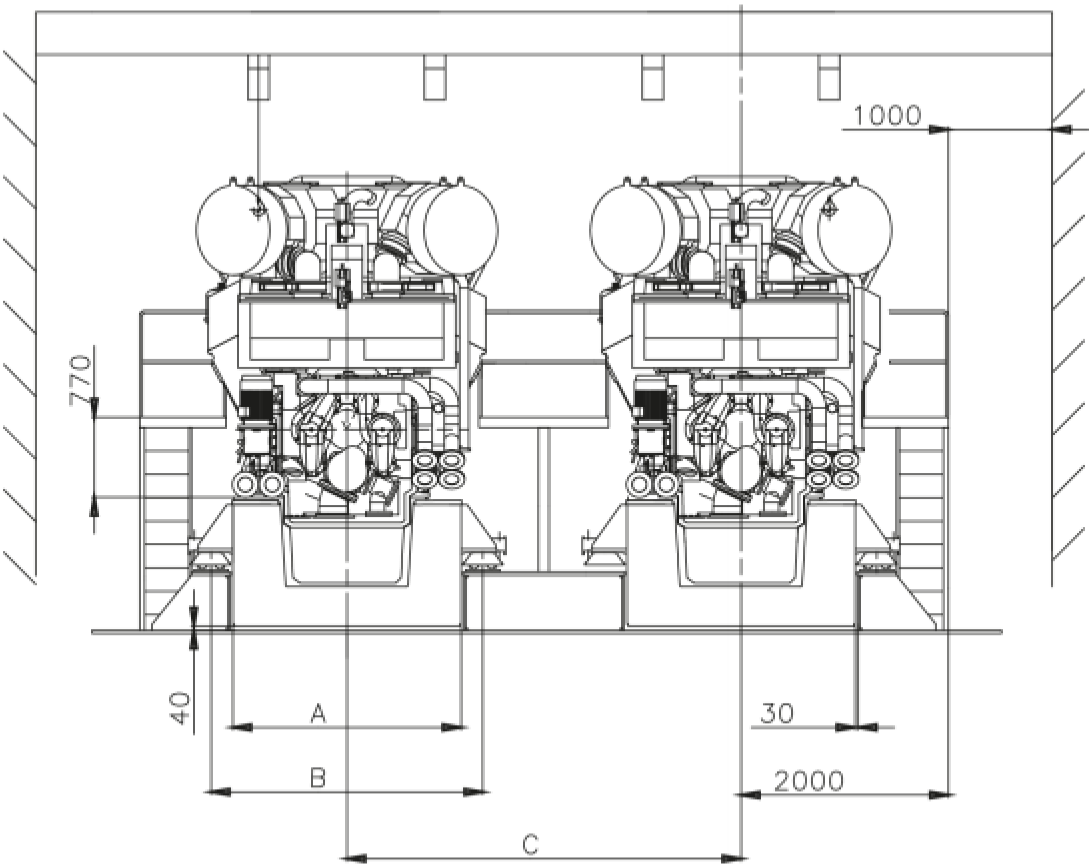
Engine	MW	LA1**	LA2**	LA3	LA4**	WA1	WA2	WA3	HA1	HA2	HA3	HA4	Weight**
W 6L34DF	480	8765	6900	1215	3160	2290	1910	1600	4000	2345	1450	1055	60
Wärtsilä 8L34DF	480	10410	8650	1285	3645	2690	2310	2000	4180	2345	1630	1055	76
W 9L34DF	480	10610	8850	1285	3845	2890	2510	2200	4180	2345	1630	1055	87
W 12V34DF	480	10260	7950	1985	3775	3060	2620	2200	4335	2120	1900	1375	99
W 16V34DF	480	11465	9130	1925	3765	3360	2920	2500	4445	2120	1850	1375	124

** Dependent on generator and flexible coupling.

All dimensions in mm. Weight in metric tons with liquids.

Figura 6-7 - Dimensiones 12V34DF

Fuente: http://cdn.wartsila.com/docs/default-source/product-files/engines/df-engine/product-guide-o-e-w34df.pdf?sfvrsn=6



Engine type	A	B	C
W 12V34DF	2200	2620	min. 3800
W 16V34DF	2200	2620	min. 3800

Figura 6-8 – Espacio libre entre diesel generadores

Fuente: http://cdn.wartsila.com/docs/default-source/product-files/engines/df-engine/product-guide-o-e-w34df.pdf?sfvrsn=6

Por lo que según nos indica el fabricante, la distancia entre generadores será 3.800 mm entre ejes.

Se pueden ver un resumen de las especificaciones a un nivel un poco más detallado ofrecido por el fabricante de nuestros diesel generadores (Wartsilla) que se adjunta en el Anexo II Documentación motor propulsor. La disposición de la planta propulsora, será mediante sistema Direct Drive (ABB) de hélices de paso fijo, en las cuales necesitaríamos dimensionar, comprar y montar dos timones.

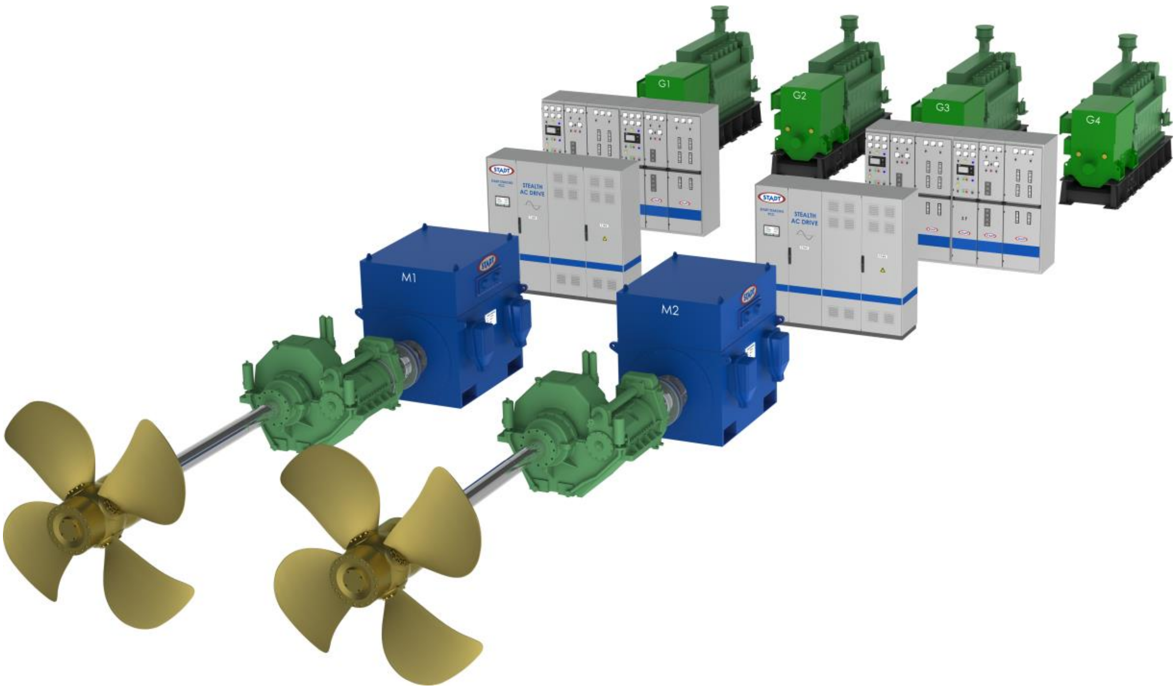


Figura 6-9 - Esquema propulsión por hélices de paso fijo

Fuente: http://www.nortrade.com/Documents/Invest/Norway%20at%20a%20glance/Norway%20abroad/STADT%20Arrangement%202012%20-%20Twin%20out%20LR.JPG

La razón por la que no hemos escogido un motor de la marca MAN, es debido que no dispone de motores dual fuel del rango de potencias que necesitamos, por lo tanto hemos tenido que buscar el motor a la casa Wartsilla.

Figura 6-10 - Rango potencias MAN

Fuente: http://powerplants.man.eu/products/dual-fuel-engines/at-a-glance

Capítulo 7. PROPULSOR

7.1. DISEÑO DEL PROPULSOR

Dos hélices de paso fijo las cuales deben estar acopladas a la reductora y a los diesel generadores.
El objetivo fundamental del proceso que se describe en este punto es la obtención de un propulsor óptimo que necesite la menor potencia posible para dotar al buque de la velocidad de servicio, la cual es 15 nudos que ha sido impuesta por la especificación del proyecto.

En el estudio de definición del propulsor han de tenerse en cuenta una serie de factores importantes como son:

- El objetivo fundamental es la consecución del máximo rendimiento posible con objeto de rebajar los costes de construcción y explotación del buque.
- No se deben presentar fenómenos de cavitación en los regímenes de velocidad en los que vaya a operar el propulsor.
- El número de palas es un factor importante que repercute en los fenómenos de cavitación, rendimiento y vibraciones en el eje y en el motor propulsor.
- La resistencia estructural tanto de hélice, eje y demás elementos del sistema propulsivo, debe ser la suficiente para evitar el riesgo de fracturas o deformaciones ante los esfuerzos a los que se vean sometidas las palas del propulsor.
- El diámetro de la hélice debe ser el máximo posible puesto que se da la relación de que a mayor diámetro mayor rendimiento del propulsor.
- Se deben cumplir ciertas disposiciones de la Sociedad de Clasificación del proyecto referentes a los huelgos entre hélice-casco-timón-línea base.

La gran mayoría de los petroleros de tamaños similares o mayores al del buque proyecto, disponen de hélices de 4 o 5 palas, puesto que las de 3 originan problemas de vibraciones y cavitación, por ello se harán todos los cálculos para propulsores de 4,5 y 6 palas respectivamente.

Para realizar este cálculo, el software necesita los datos del motor que hemos elegido, se introducen en el programa, y se escoge “Mode: Propulsion”.

En las siguientes páginas se recogen los cálculos llevados a cabo para determinar las características de nuestros propulsores mediante el software informático NavCAD.
Realizamos los cálculos mediante el NavCAD a fin de poder realizar una comparación con nuestros propulsores diseñados manualmente.

Dichos cálculos se corresponden a las hélices de 4, 5 y 6 palas. El objetivo final será comprobar cuáles de estas dos tendrá un comportamiento más idóneo tras haber obtenido las características geométricas de cada una de ellas.

Para ello será esencial realizar un estudio de cavitación a cada una de ellas, el cual se realizará por el método de Keller, para obtener así las características óptimas de dichas hélices, ya que los criterios de cavitación serán los que en última instancia fijen las características geométricas finales de las hélices a proyectar.

El proceso que se ha llevado a cabo es el siguiente:

Primeramente es necesario determinar el número de propulsores que llevará instalado el buque. Aunque la mayoría de buques similares al nuestro portan una única línea de ejes, como sabemos es requisito necesario por parte del armador el que nuestro buque posea dos líneas de ejes, portando así dos propulsores. Aunque esto es algo atípico en este tipo de buques, como es un requisito imprescindible se diseñará el buque atendiendo a dicho requerimiento, ya que el armador así lo desea.

Por otro lado, se ha concretado que en este tipo de buques lo más común es utilización de hélices de paso fijo ya que las condiciones de navegación no hacen necesario la utilización de hélices de paso controlable, que además son más caras. Desde el punto de vista del armador, teniendo en cuenta que la instalación de una hélice de paso controlable no supone una ventaja adicional relevante y el aumento del costo que supondría, no se ve justificable en ningún caso la instalación de una de estas hélices. Por tanto queda fijada la utilización de unas hélices de paso fijo.

7.1.1. DATOS INICIALES

Lo primero que tenemos que hacer es introducir nuestros datos al programa, iremos mostrando mediante capturas de pantalla como lo vamos haciendo para que así quede constancia de que datos le introducimos al programa para la realización del cálculo.
Lo primero que definiremos es las propiedades del fluido en nuestro caso el agua del mar:

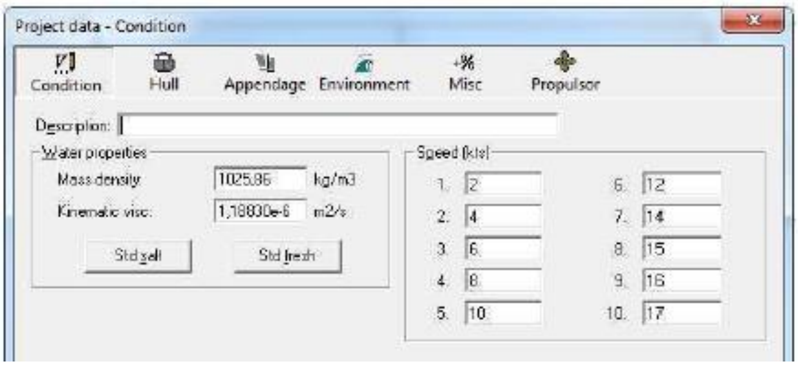


Figura 7-1 - Fluido hélice
Fuente: NavCAD

Lo siguiente que haremos será definir el margen de mar a aplicar que como sabemos será del 10%:

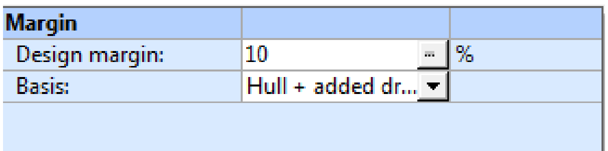


Figura 7-2 - Margen de mar
Fuente: NavCAD

Para calcular el valor final, se procederá al dimensionamiento del propulsor con algunas variaciones en NavCAD:

- Dimensionamiento del propulsor: será la opción de *"By Power".* Ya que ahora el dato de la potencia es conocido.
- Propeller series: B Series
- Numero de palas: 4, 5 y 6

- Se fija la opción variable de rpm.

Propulsor		
Count:	2	
Propulsor type:	Propeller series	
Propeller type:	FPP	
Propeller series:	B Series	
Propeller sizing:	By power	

Figura 7-3 - Datos hélice
Fuente: NavCAD

También en la ventana de *propeller sizing* se marcarán las opciones:
• Se varía la velocidad de diseño
• Se especifica como potencia de referencia la del motor introducido (9.000 kW)

El resto de valores serán dimensionados por el programa.

Propeller sizing

To size			
Gear ratio:	Keep	1,00	
Expanded area ratio:	Size	**0,518**	
Propeller diameter:	Keep	7000,0	mm
Propeller mean pitch:	Size	**4576,7**	mm
Design condition			
Design speed:		15,00	kt
Reference power:		**9000**	kW
Design point:		1,000	
Reference RPM:		125,0	
Design point:		1,030	
Max prop diam:		8000,0	mm
Review			
Tip speed:		47,19	m/s

Size | Save report | OK | Cancel | Help

Figura 7-4 - Características generales hélice
Fuente: NavCAD

7.2. COMPARATIVA Y ELECCIÓN DEL NÚMERO DE PALAS

7.2.1. PROPULSOR DE 3 PALAS

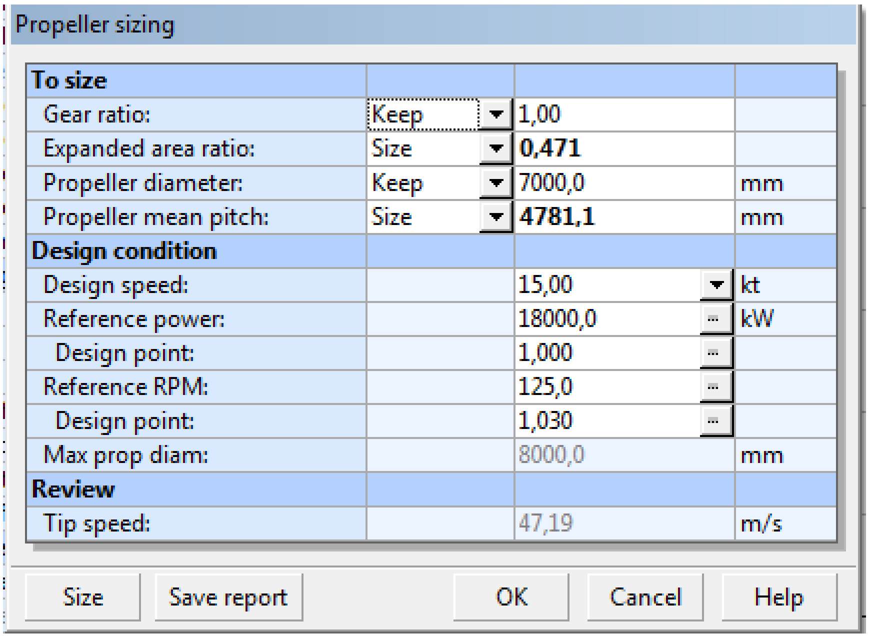


Figura 7-5 - Características hélice de 3 palas
Fuente: NavCAD

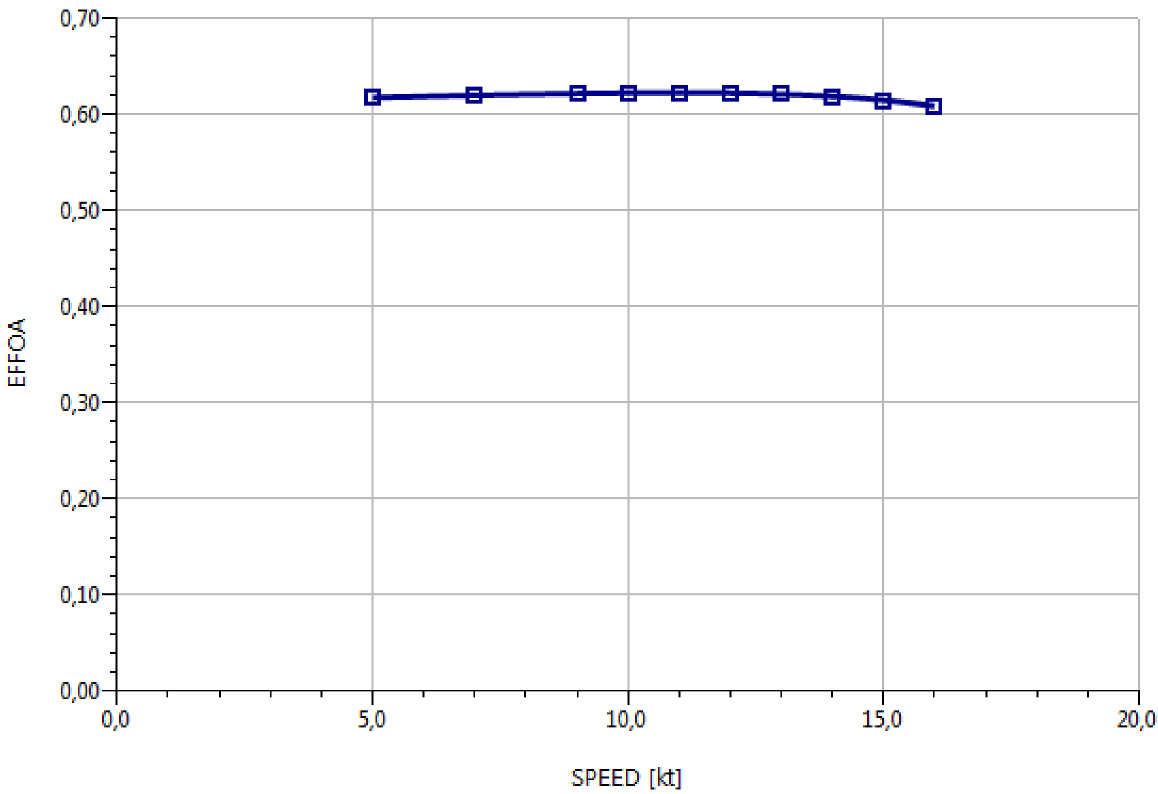


Figura 7-6 - Rendimiento hélice de 3 palas
Fuente: NavCAD

7.2.2. PROPULSOR DE 4 PALAS

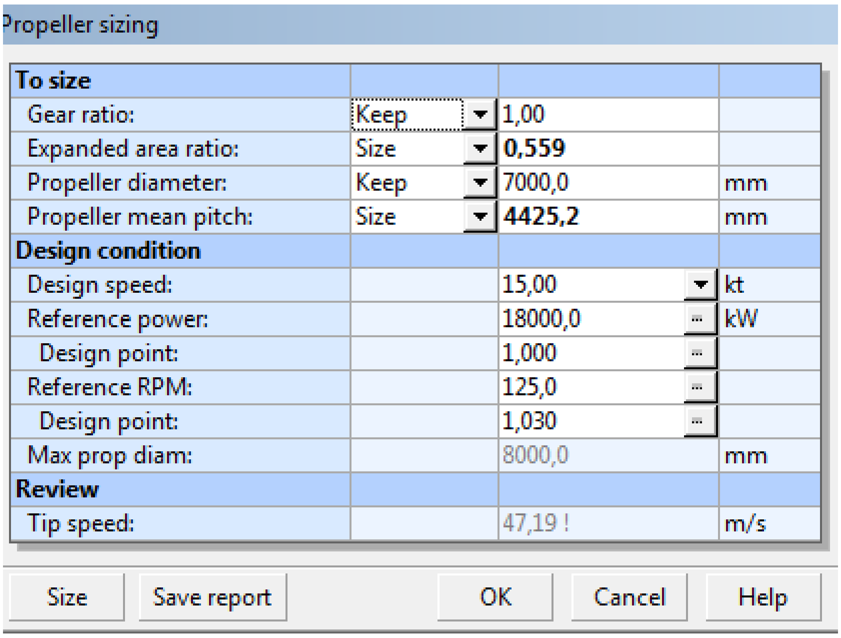


Figura 7-7 - Características hélice de 4 palas
Fuente: NavCAD

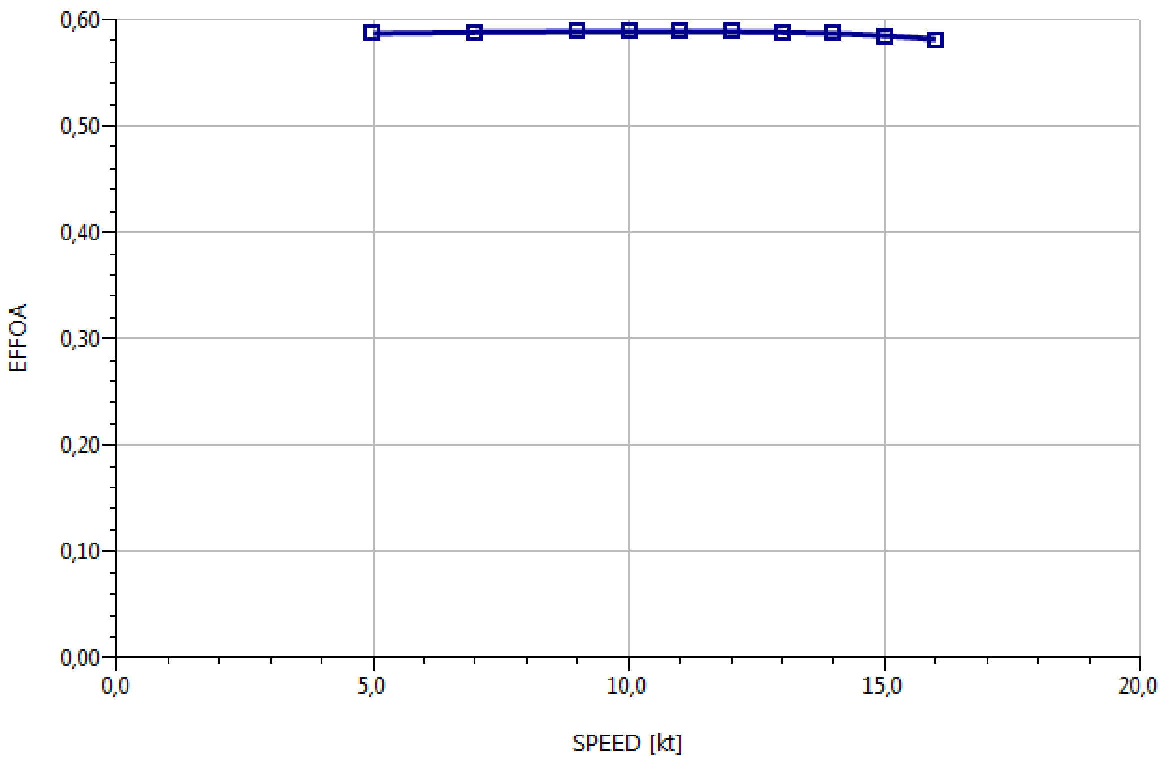


Figura 7-8 - Rendimiento hélice de 4 palas
Fuente: NavCAD

7.2.3. PROPULSOR DE 5 PALAS

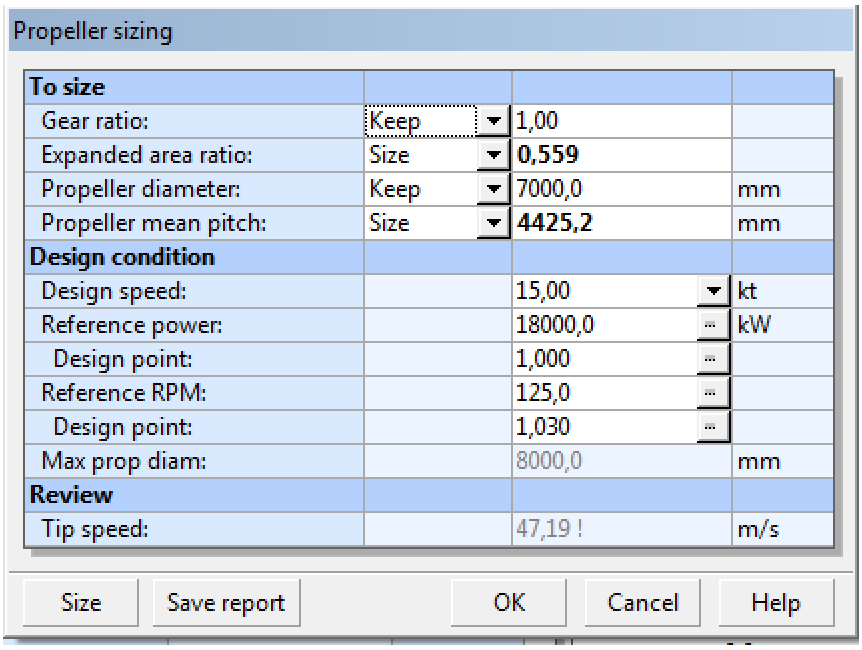


Figura 7-9 - Características hélice de 5 palas
Fuente: NavCAD

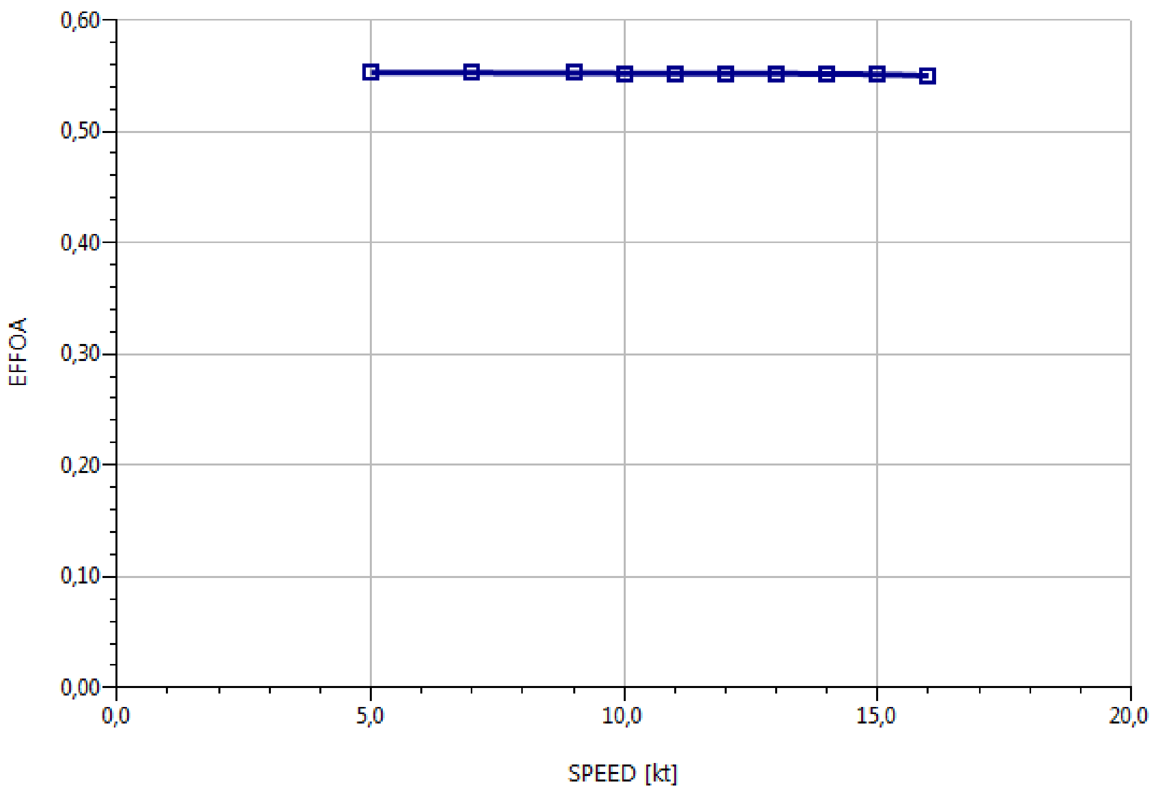


Figura 7-10 - Rendimiento hélice de 5 palas
Fuente: NavCAD

7.2.4. PROPULSOR DE 6 PALAS

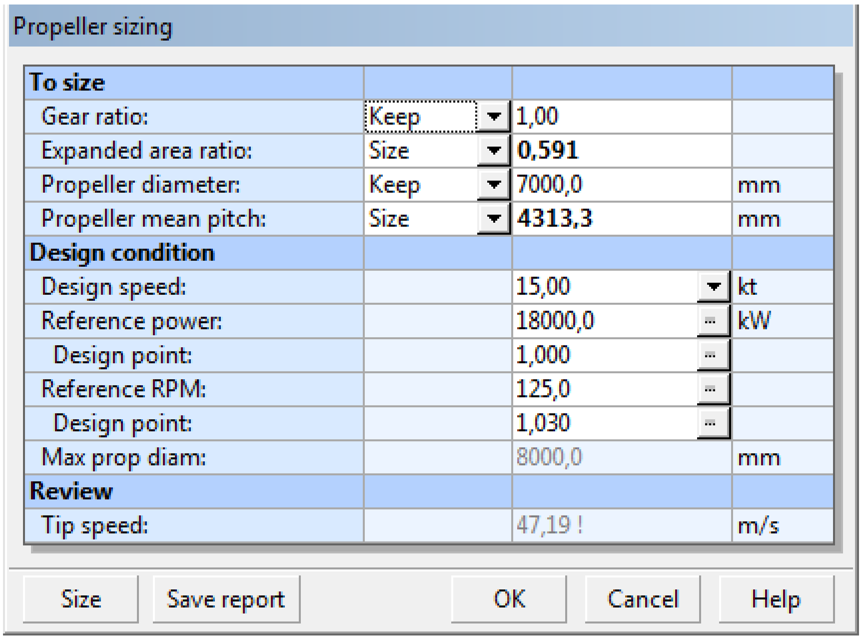


Figura 7-11 - Características hélice de 6 palas
Fuente: NavCAD

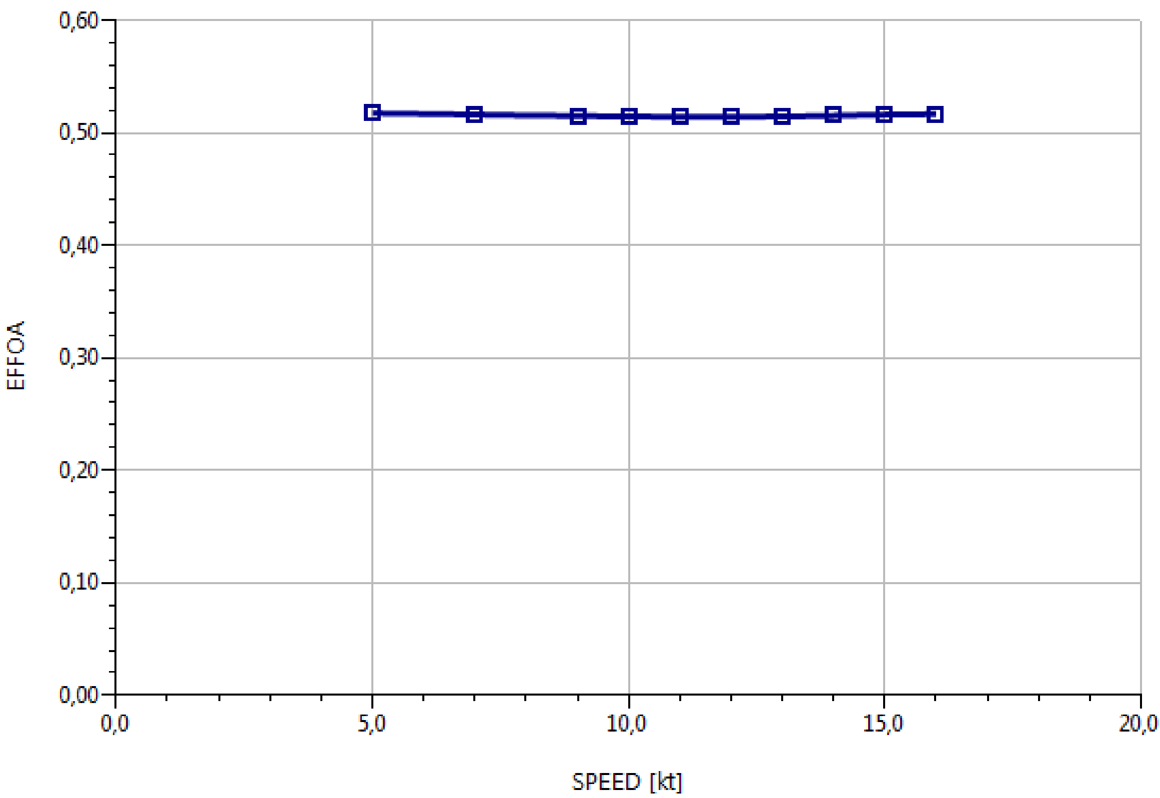


Figura 7-12 - Rendimiento hélice de 6 palas
Fuente: NavCAD

7.2.5. PROPULSOR ELEGIDO

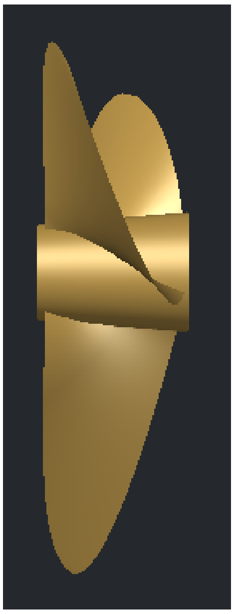
El resumen de datos obtenidos para 4, 5 y 6 palas es el siguiente:

Nº de Palas	Rendimiento	Diámetro (mm)	Pitch	Ad/Ao	Cavitación (%)
3	**0,6147**	**7.000**	**4.781,1**	**0,471**	**2,0**
4	0,5853	7.000	4.577,6	0,518	2,0
5	0,5524	7.000	4.425,2	0,559	2,0
6	0,5161	7.000	4.313,3	0,591	2,0

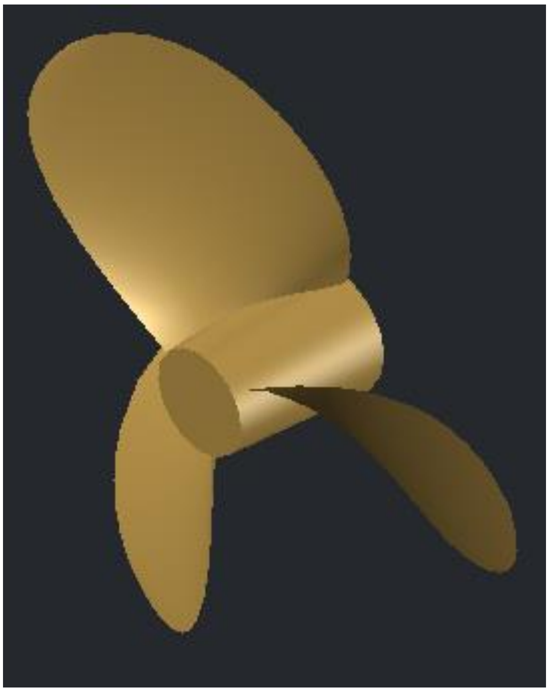
Tabla 7-1 - Resumen características de propulsores
Fuente: Propia

La hélice elegida tendrá 3 palas con un rendimiento de 0,6147, ya que de las posibles se escoge la de mayor rendimiento.

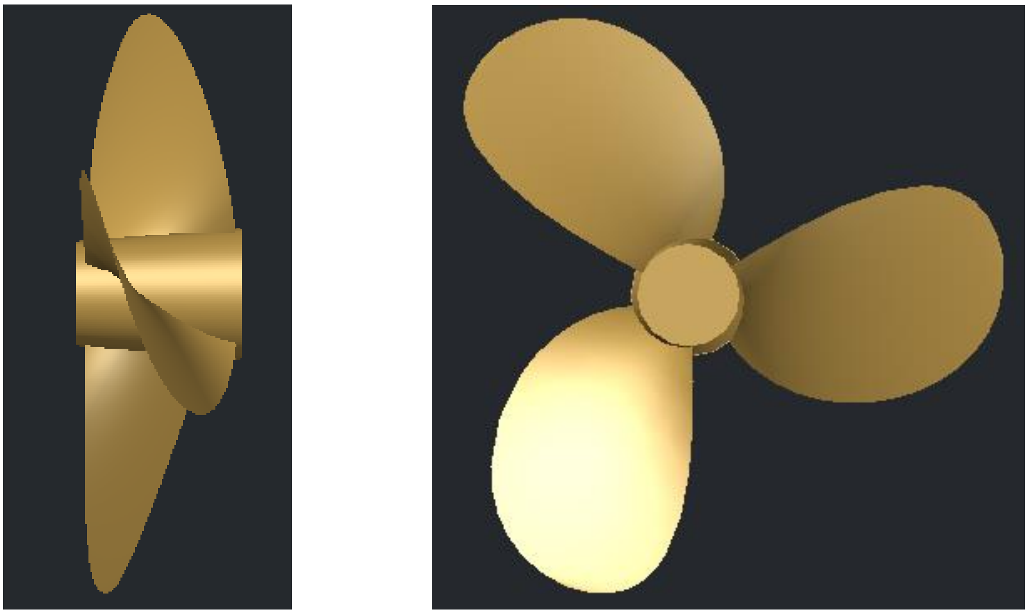
A continuación, mediante las herramientas del software NavCAD, exportamos las formas de nuestra hélice en formato .ign, pudiendo importarlas mediante el software AutoCAD.
Finalmente, mostramos las vistas del propulsor dimensionado que hemos exportado:



Hélice vista en planta



Hélice vista en perspectiva



Hélice vista de perfil　　Hélice vista en alzado

Figura 7-13 - Propulsor dimensionado de 4 palas
Fuente: NavCAD

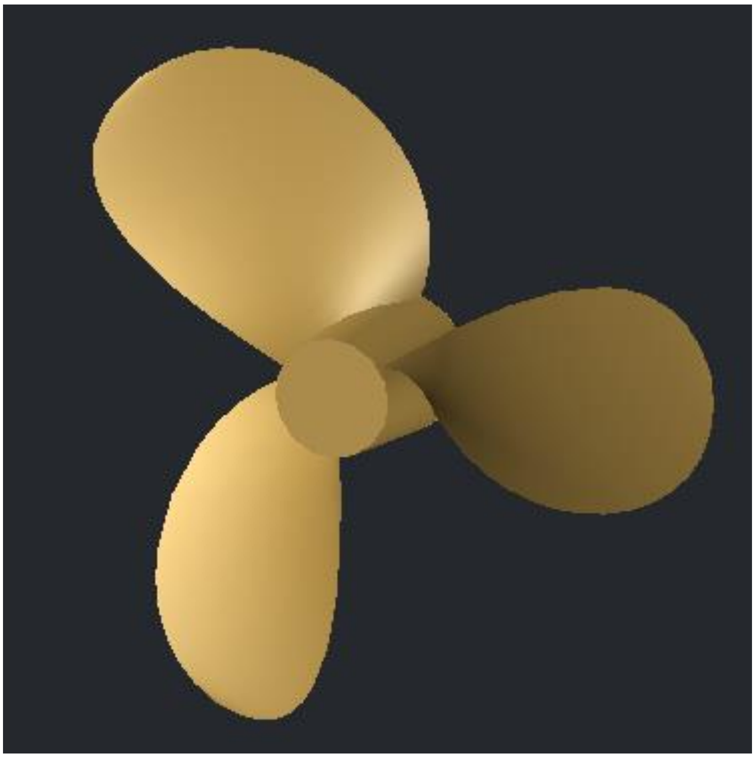


Figura 7-14 – Renderizado de hélice
Fuente: Propia

7.3. HUELGOS DE LA HÉLICE

A la hora de seleccionar el diámetro máximo, se tendrán en cuenta los huelgos entre la hélice y el casco, que ejercen gran influencia sobre la excitación producida por las fluctuaciones del par y del empuje. Estas claras vienen definidas por la sociedad de clasificación (Lloyd's Register en nuestro caso) en la parte 3, capítulo 6, sección 7.

Table 6.7.3 Recommended propeller/hull clearances

Number of blades	Hull clearances for single screw, in metres, *see* Fig. 6.7.7(a)				Hull clearances for twin screw, in metres, *see* Fig. 6.7.7(b)	
	a	*b*	*c*	*d*	*e*	*f*
3	1,20$K\delta$	1,80$K\delta$	0,12δ	0,03δ	1,20$K\delta$	1,20$K\delta$
4	1,00$K\delta$	1,50$K\delta$	0,12δ	0,03δ	1,00$K\delta$	1,20$K\delta$
5	0,85$K\delta$	1,275$K\delta$	0,12δ	0,03δ	0,85$K\delta$	0,85$K\delta$
6	0,75$K\delta$	1,125$K\delta$	0,12δ	0,03δ	0,75$K\delta$	0,75$K\delta$
Minimum value	0,10δ	0,15δ	t_R	—	3 and 4 blades, 0,20δ 5 and 6 blades, 0,16δ	0,15δ

Symbols

L as defined in 1.4.1

C_b = moulded block coefficient at load draught

$$K = \left(0{,}1 + \frac{L}{3050}\right)\left(\frac{3{,}48 C_b P}{L^2} + 0{,}3\right)$$

$$\left(K = \left(0{,}1 + \frac{L}{3050}\right)\left(\frac{2{,}56 C_b P}{L^2} + 0{,}3\right)\right)$$

t_R = thickness of rudder, in metres, measured at 0,7Rp above the shaft centreline

P = designed power on one shaft, in kW (shp)

R_P = propeller radius, in metres

δ = propeller diameter, in metres

NOTE
The above recommended minimum clearances also apply to semi-spade type rudders.

Figura 7-15 - Dimensiones de los huelgos

Fuente: Lloyd's Register Of Shipping

$$K = \left(0{,}1 + \frac{L}{3050}\right) * \left(\frac{2{,}56 * Cb * P}{L^2} + 0{,}3\right)$$

$$K = 0{,}120$$

$$K = \left(0{,}1 + \frac{L}{3050}\right) * \left(\frac{3{,}48 * Cb * P}{L^2} + 0{,}3\right)$$

$$K = 0{,}145$$

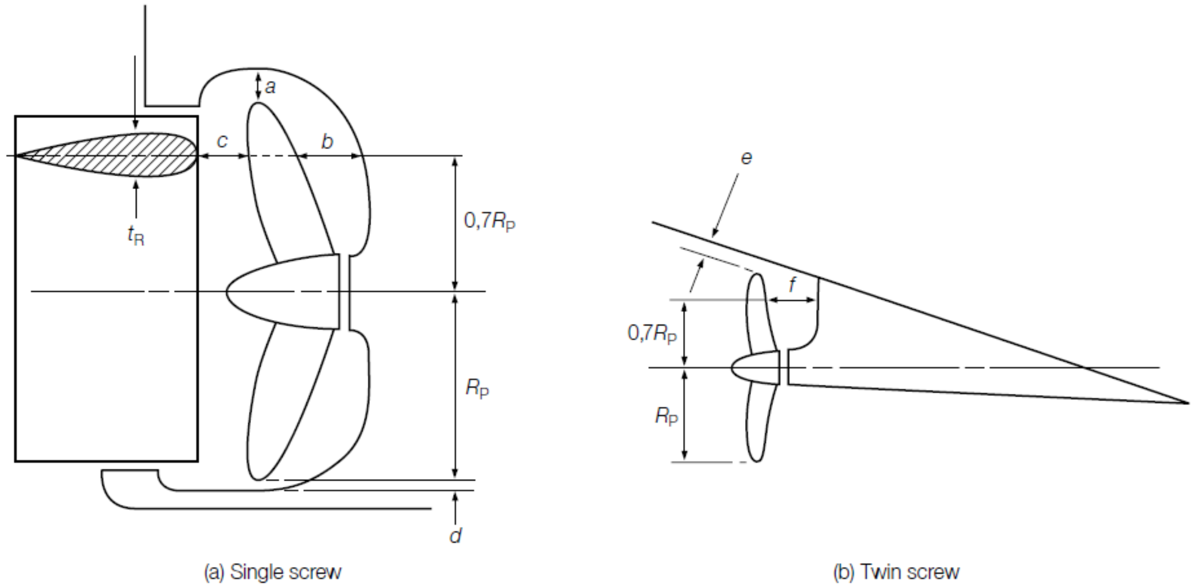


Fig. 6.7.7 Propeller clearances

Figura 7-16 - Esquema de los huelgos

Fuente: Lloyd's Register Of Shipping

A continuación calcularemos los huelgos mínimos recomendados para la hélice de nuestro buque, que ya mencionamos con todo lujo de detalles en el presente cuaderno que será de 4 palas y tendrá 7 metros de diámetro aproximadamente.

Para estos valores de la hélice, los valores mínimos serán los siguientes

HUELGOS DE LA HÉLICE					
Concepto	Valor mínimo L.R.S		Holguras casco L.R.S		Holgura real
	Z=3	Z=4	Z=3	Z=4	
k (m)	0,145				
e (m)	1,4	1,4	1,218	1,015	2,255
f (m)	1,05	1,05	1,218	1,218	1,876

Tabla 7-2 - Resumen huelgos mínimos
Fuente: Propia

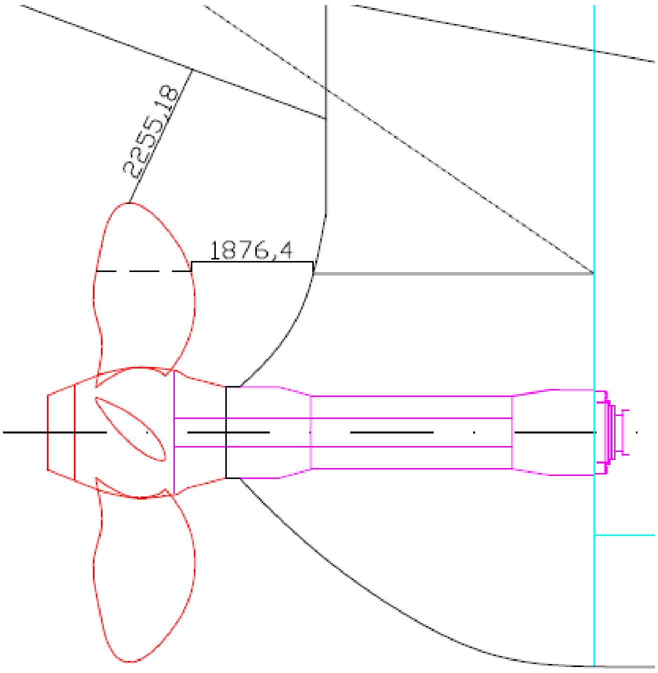


Figura 7-17 - Huelgos de nuestra hélice
Fuente: Propia

Como podemos observar cumplimos holgadamente las claras mínimas impuestas por la sociedad de clasificación, por lo tanto la configuración de nuestros elementos en el codaste de nuestro buque es correcta.

7.4. MATERIAL DE LA HÉLICE

Seleccionamos como material de la hélice una aleación de níquel, aluminio y bronce, al igual que la hélice de nuestro buque base.

Capítulo 8. TIMÓN

Los timones son los aparatos de gobierno que dotan de maniobrabilidad y rumbo al buque.

La maniobrabilidad del buque es la característica del mismo que le permite responder adecuadamente ante determinadas adversidades.

8.1. ÁREA MÍNIMA

Se calculará el valor del área según el libro "Proyecto básico del buque mercante". Se estimara como primera aproximación:

$$Lpp * T = 33 \text{ m2}$$

LPP*T (entre un 1,5 y un 2,5 %)	
49,5 m2	82,5 m2

Tabla 8-1 - Primera estimación del timón

Fuente propia

La segunda estimación que realizaremos, será la de la sociedad de clasificación Det Norske Veritas. El Área del timón no debe ser menor que:

$$AR = 0{,}01 * LPP * T * (1 + 50 * Cb^2 * \left(\frac{B}{LPP}\right)^2)$$

$$AR = 60.939 \approx 61\ m^2$$

Para terminar aplicamos una fórmula propuesta por Japón en IMO:

$$AR = 0{,}01 * LPP * T * (k_1 * \frac{B}{LWL * Cb} + k_2)$$

Siendo:

- k1 = 54 / (7,2 - 30 x V / LPP)
- k2 = 0,0008 x B / T [LPP / (B x Cb)]2
- V: la velocidad de servicio en nudos

$k_1 = 10{,}4761$

$k_2 = 0{,}1070$

$AR = 66{,}947\ m^2$

Por lo que el área de timón final para nuestro buque será de 61 metros cuadrados.

La relación de aspecto de un timón es el cociente entre la altura y la longitud media del timón. Es conveniente que sea próxima a 1,5.

La cuerda del timón puede calcularse fácilmente a partir de la altura y el área del timón que ya han sido calculados puesto que el área es el producto de la altura por la cuerda. Por lo tanto:

$$c = \frac{Ar}{h} = 6{,}4\ m$$

La relación de alargamiento es el cociente entre la altura y la cuerda del timón por lo que su valor ya puede calcularse según la expresión:

$$\lambda = \frac{h}{c} = 1{,}5$$

Para tratar de conseguir un timón lo menos pesado posible y por tanto exigir una menor potencia del servo, se elige una relación de espesor de 0,15 que cumple con seguridad la condición de que el ángulo de desprendimiento de flujo sea superior a 35º.

Por lo tanto, si E = 0,14 ya puede calcularse el valor del espesor, puesto que se dispone del valor de la relación entre el espesor y la cuerda. Así pues:

$$t = E * c = 0{,}14 * 6{,}4 = 0{,}896\ m$$

Area timón	Altura timón	Cuerda del timón	Parte compensada
61,44	9,6	6,4	1,5

Tabla 8-2 - Características del timón

Fuente: Propia

8.2. FUERZA SOBRE EL TIMÓN

A continuación se calcula la fuerza en las condiciones de avante y ciando mediante la fórmula:

$$F = 0{,}044 * k_1 * k_2 * k_3 * AV^2$$

Dónde:

- F: fuerza sobre el timón.
- A=94,11 m^2; área de pala.
- V=15 knot; velocidad máxima de servicio con el barco en la línea de flotación de carga de verano.
- k1: es un coeficiente que toma el valor 1,1 para perfiles NACA.
- k2 toma el valor 1 en general.
- k3:

$$k_3 = \frac{H^2}{A_t} + 2 \leq 4$$

Dónde:

H: LA altura media del timón, 9,6 metros

$$k_3 = 3{,}5 \leq 4$$

La fuerza sobre el timón avante será:

$$F = 2.341{,}7856\ kN$$

La fuerza sobre el timón ciando será:

$$F = 585{,}44\ kN$$

8.3. TIPO DE TIMÓN

Lo primero que vamos a ver en este apartado del capítulo es ver los tres tipos de timón más usuales que se suelen montar y elegir por cuál de ellos nos decantamos:

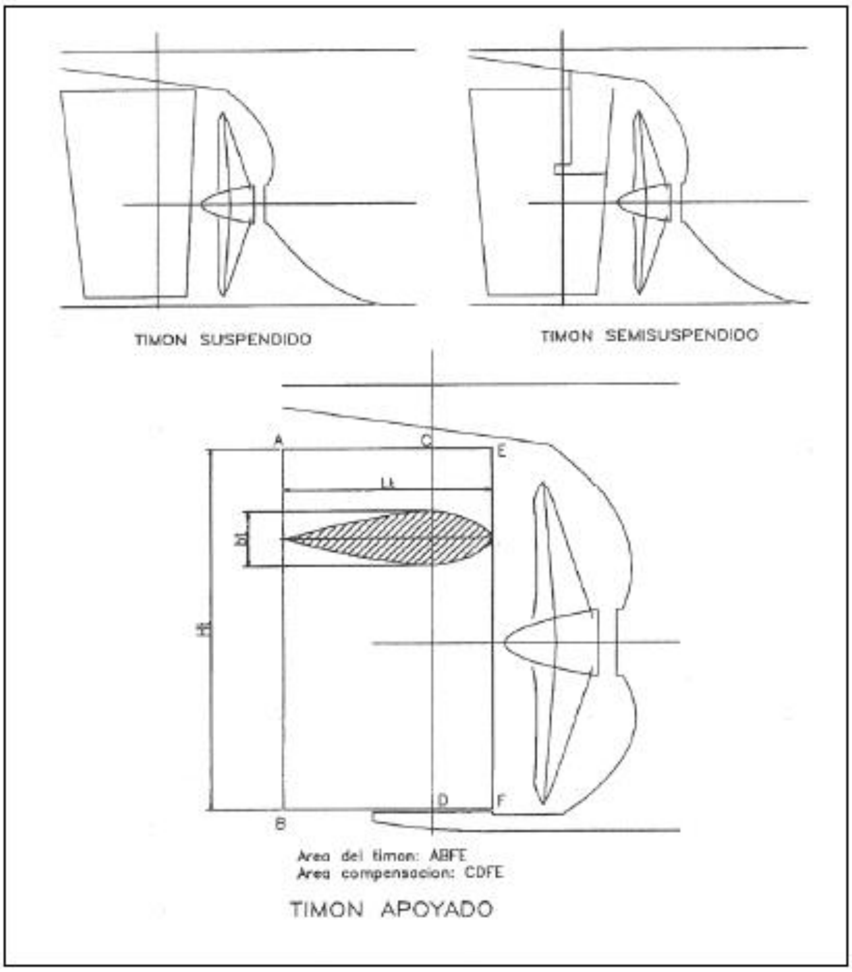


Figura 8-1 - Tipos de timón
Fuente:http://www.japanham.com/wp/wp-content/themes/japanham/en/img/service/schilling/img4.jpg

Una vez visto los tres tipos de timones más usuales, nos decantamos dentro de los tipos de timón que podemos encontrarnos en el mercado, por un timón de perfil tipo NACA00 con bulbo, que nos proporcionará a la velocidad de servicio de nuestro buque, 15 nudos, la mayor eficiencia en comparación con el resto de velocidades y el resto de los timones, una serie de ventajas que explicaremos a más adelante.

Figura 8-2 - Timón Mariner con bulbo
Fuente: http://www.wartsila.com/file/Wartsila/en/1270037667661a1267106724867-wartsila-o-rudders.pdf

Este timón, reduce el ángulo de ataque local en los timones de borde de ataque, esto le da al timón más eficiencia debido a una menor fricción y una mejor recuperación de la energía de rotación de la estela de la hélice.

A continuación, enumeraremos una serie de ventajas que poseen estos timones en comparación con los convencionales, al final del documento, en el Anexo III Documentación timones de rendimiento, podremos encontrar toda la información disponible acerca de estos timones, sus características y sus mejoras tanto en eficiencia como en reducción de combustible y potencia.

***Ventajas*[1]**

- Un Diseño convencional de construcción, junto con una instalación sencilla
- Hélice y timón están diseñados como una sola unidad para la eficiencia de propulsión óptima.
- Eficiencia de propulsión se incrementa en un rango del 3-6%.
- Mejora la maniobrabilidad a baja velocidad.
- Menor nivel de ruido y de las vibraciones de la hélice.
- Bajo mantenimiento
- El diseño es simple y robusto.
- Casi tan fácil de instalar como un sistema de hélice-timón convencional.
- Los mejores resultados se obtienen en los buques de un solo eje con un coeficiente de bloque de 0,75-0,85 y con una velocidad en el rango de 14 a 16 nudos. Aquí la ganancia de eficiencia puede ser de un 6-9% en comparación con las soluciones convencionales.

[1] http://www.rolls-royce.com/marine/products/propulsors/promas/

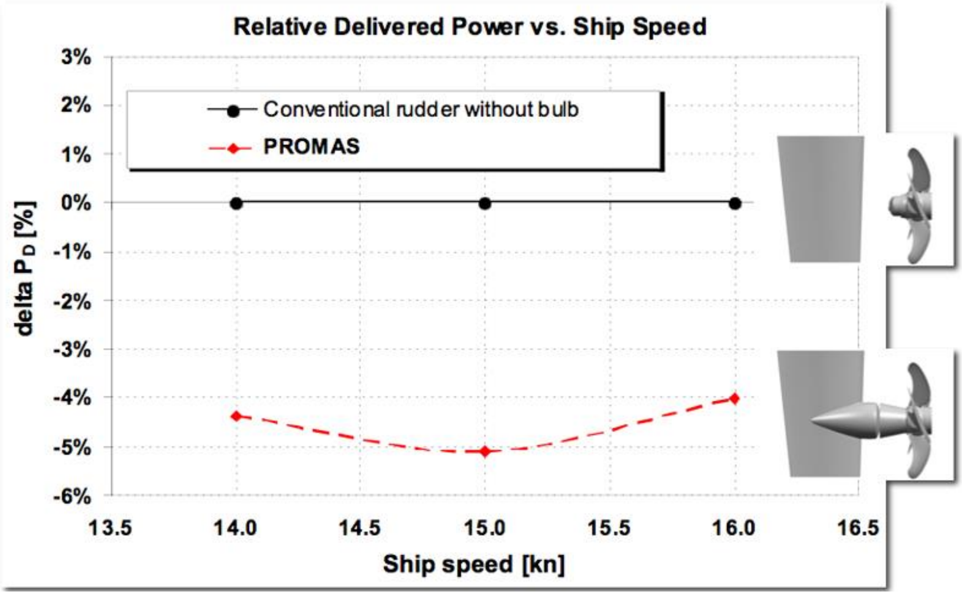


Figura 8-3 - Ahorro de potencia
Fuente: http://www.rolls-royce.com/marine/products/propulsors/promas/

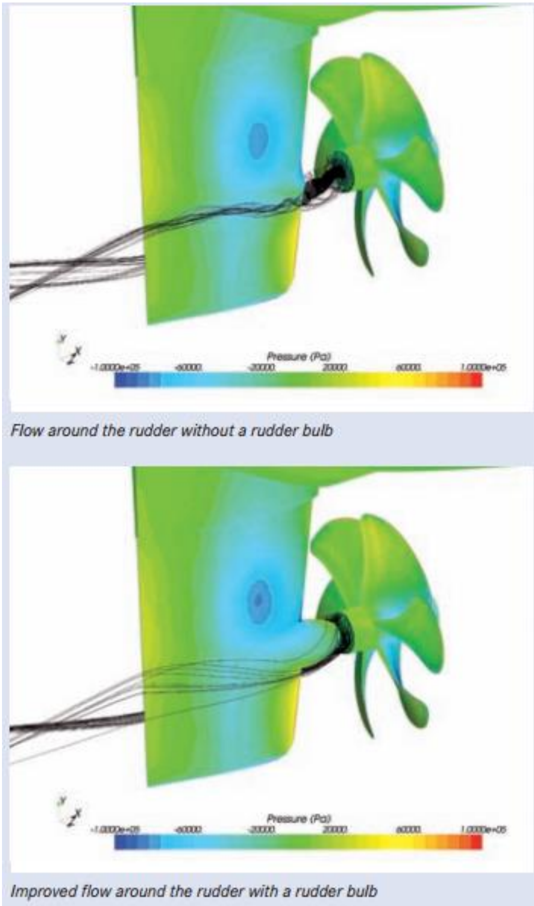


Figura 8-4 - Comparativa de timones mediante CFD
Fuente: http://www.becker-marine-systems.com/05_downloads/zzpdf/product_pdf/Becker_Bulb_Rudders.pdf

Observamos que el sistema hélice-timón nos proporciona grandes ventajas, como el ahorro de potencia y la maniobrabilidad. Habrá que estudiar más adelante si realmente compensa este sistema, debido a la inversión inicial a realizar.

Por lo tanto, nuestro buque quedará con la siguiente disposición de elementos, 2 hélices de paso fijo, unidos a dos timones con bulbo, de tipo Mariner de la marca Wartsilla o Promas de la marca Roll Royce, nuestro buque quedara de la siguiente manera, en la cual podemos apreciar en la imagen:

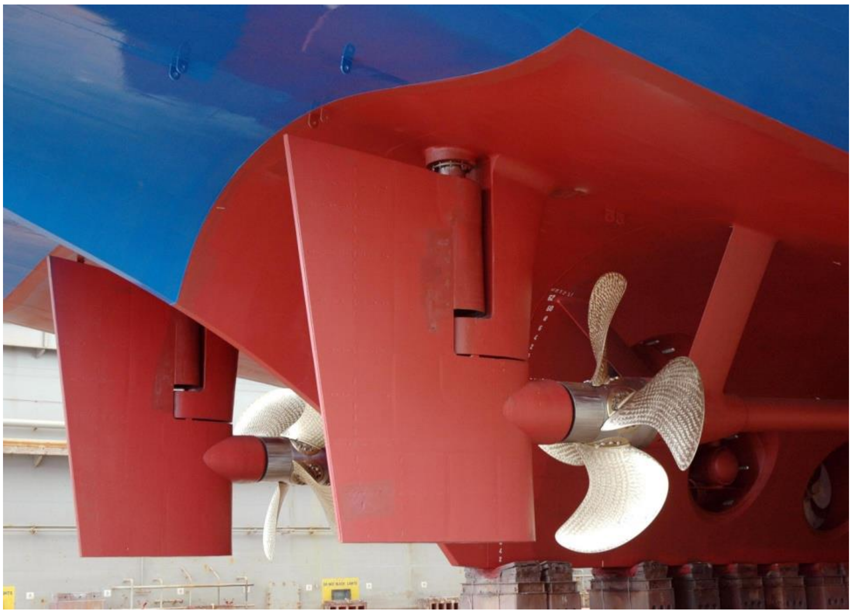


Figura 8-5 - Dos líneas de ejes con timones Promas
Fuente: http://www.motorship.com/__data/assets/image/0038/417899/Rolls-Royce-Promas-Lite.jpg

En el Anexo I Plano línea de ejes, podremos ver completamente, las dos líneas de ejes, el codaste modificado para la entrada de los dos ejes, así como la disposición de los elementos y su distribución en el barco.

Capítulo 9. CALCULO DE LA MANIOBRABILIDAD

A continuación estimaremos mediante fórmulas y gráficos una serie de características propias de nuestro buque, que deberán superar los criterios de maniobrabilidad de la IMO. En este apartado se estudiaran los siguientes conceptos:

9.1. FACILIDAD DE EVOLUCIÓN (TURNING ABILITY)

Esta característica nos proporciona una idea del espacio que necesita el buque para realizar grandes cambios de rumbo.

Viene definido por los siguientes parámetros:

9.1.1. DIÁMETRO DE GIRO (TURNING DIAMETER)

Se obtiene mediante la siguiente expresión:

Para buques de dos hélices:

3.6.1.1.1 - Diámetro de giro (turning diameter)

a) Buques de 1 hélice al calado de proyecto

DG = LPP [4,19 - 203 CB / DELR + 47,4 TRI / LPP - 13 B / LPP +
+ 194 / DELR - 35,8 AR / (LPP × T) + 7,79 AB / (LPP × T)] **(3.6.1)**

Figura 9-1 - Diámetro de giro

El proyecto básico del buque mercante

Esta fórmula se refiere a codastes abiertos, sin considerar diferencias entre los giros a babor y estribor; si el codaste es cerrado, el 6º término, -35,8 AR / (LPP × T), se debe cambiar por + 3,82 AR / (LPP × T)

Ejemplo 1:

LPP = 170 m	B = 27 m	T = 10 m	TRI = 0	CB = 0,66
AR = 29 m²	AB = 28 m²	V = 19,5 nudos		

Con un ángulo de timón DELR de 35°, el diámetro de giro sería:

a) Con codaste abierto DG = 570,8 m (3,35 LPP)

b) id cerrado DG = 685,7 m (4,03 LPP)

Figura 9-2 - Diámetro de giro variables

El proyecto básico del buque mercante

$$Dg = LWL * \left(4{,}19 - \frac{203CB}{DELR} + \frac{47{,}4TRI}{LWL} - \frac{13B}{LWL} + \frac{194}{DELR} - \frac{35{,}8AR}{LWL * T} + \frac{7{,}79AB}{LWL * T}\right)$$

Siendo:

DELR
35
TRI
2
Ab
12,43

Tabla 9-1 - Variables del diámetro de giro

Fuente propia

$$DG = 493{,}029\ metros$$

Es decir, 2,41 veces la LPP de nuestro buque

IMO exige que DG < 4,641*Lpp por lo que cumplimos la norma.

9.2. Diámetro táctico o de evolución:

Esta expresión no debe ser mayor que 5LPP por la IMO

$$DT = LPP * \left(0{,}91 * \frac{DG}{LPP} + 0{,}234 * \frac{V}{LPP^{0.5}} + 0{,}675\right)$$

$$DT = 649{,}218\ \text{m} = 2{,}950\ \text{Lpp}$$

IMO requiere que el diámetro táctico o de evolución (DT) sea menor a 6.115 x Lpp:

Por lo que observamos que cumple de creces con la normativa de la IMO

9.3. Avance (Advance)

Para cumplir con la norma IMO, este valor no excederá de 4,5 LPP

$$ADVC = LPP * \left(0{,}519 * \frac{DT}{LPP} + 1{,}33\right)$$

$$ADVC = \quad 629{,}544\ \text{m} = 2{,}861\ \text{Lpp}$$

9.4. Caída o transferencia (transfer)

Se calcula mediante la siguiente expresión:

$$TRANS = Lpp * \left(\frac{0{,}497DT}{Lpp} - 0{,}065\right) = 308{,}365\ \text{m}$$

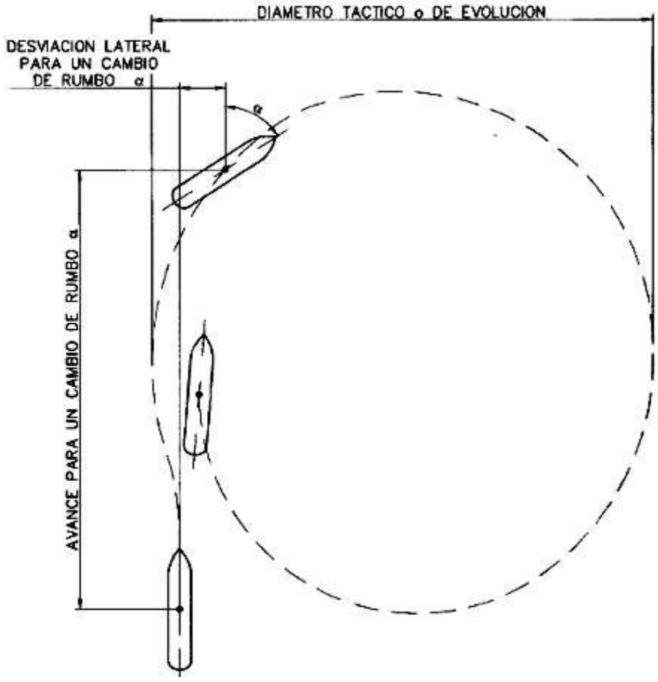


Figura 9-3 - Circulo de evolución

El proyecto básico del buque mercante

9.5. FACILIDAD PARA MANTENER EL RUMBO (COURSE KEEPING ABILITY)

Petroleros con primer ángulo de rebasamiento en la maniobra en Z de 10º/10º

$$\frac{DELO}{DELR} = 3{,}2 * \left(CB * \frac{B}{LPP} + 0{,}10\right) = 0{,}769$$

- En la maniobra de zig-zag en 10º/10º el valor del primer ángulo de rebasamiento no excederá de:
  - 10º, si la relación LPP/V es menor de 10 segs. Con la eslora y la velocidad en m y m/s, respectivamente.
  - 20º, si la relación LPP/V es mayor de 30 segundos.
  - [5 + 0,5 × (LPP/V)] grados, si LPP/V está entre 10s y 30s.
- En la maniobra de zig-zag en 10º/10º el valor del segundo ángulo de rebasamiento no excederá de los valores anteriores en más de 15º.
- En la maniobra de zig-zag en 20º/20º el valor del primer ángulo de rebasamiento no excederá de 25º.

Figura 9-4 - Maniobra zig-zag

Fuente: El proyecto básico del buque mercante

Petroleros con primer ángulo de rebasamiento en la maniobra en Z de 20º/20º

$$\frac{DELO}{DELR} = 5{,}20 * \left(CB * \left(\frac{B}{LPP}\right) + 0{,}019\right)$$

$$DELO/DELR = 0.77546$$

LPP/V (m/s)
28,50971922
Grados
19,25485961
DELO
15,50923636
1,2DELO
18,61108364

Tabla 9-2 - Variables de la maniobra zig-zag

Fuente: Propia

9.6. FACILIDAD DE PARADA (STOPPING ABILITY)

Este apartado indica la distancia que recorrerá el buque tras una maniobra de todo atrás.

Con este cálculo queremos conocer la distancia máxima para prevenir colisiones. Según la normativa IMO, el valor de la distancia obtenida no excederá de 15 veces la eslora entre perpendiculares del petrolero.

$$PP = 0{,}305 * V^3 * \frac{DISW}{PBA * DP}$$

Donde PBA es la potencia del buque en retroceso, la cual se estima en un 35% de su capacidad:

$$PBA = 13858{,}44 * 0{,}35 = 4.655\ Kw$$

Teniendo PBA, calculamos PP.

$$PP = 2.527{,}236\ \mathrm{Kw}$$

Por último, obtendremos la distancia buscada.

$$RH = 0{,}305 * e^{0{,}773 - 5*10^{-5}*PP*0{,}617*\ln(PP)} * DISW^{\frac{1}{3}}$$

$$RH = 3.154{,}474\ m$$

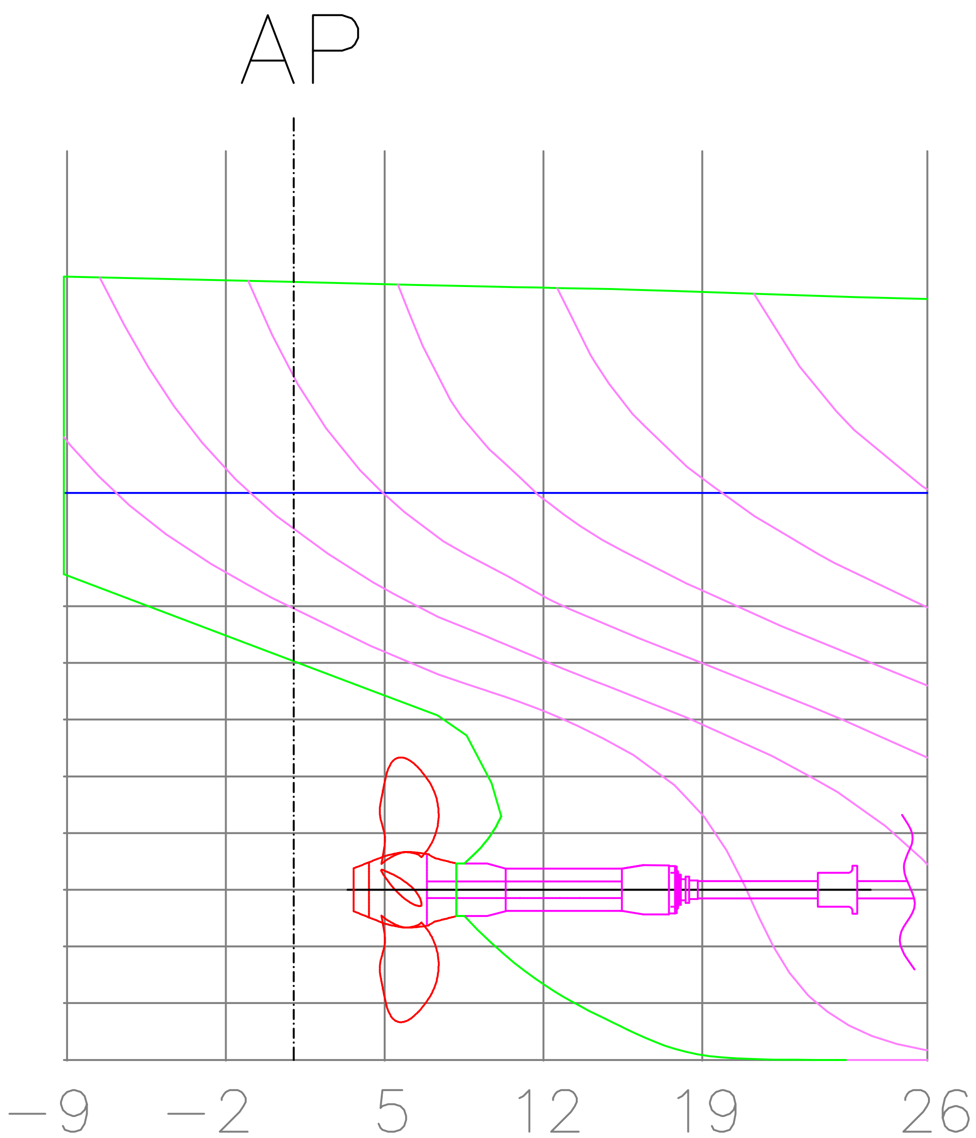
Siendo el máximo de:

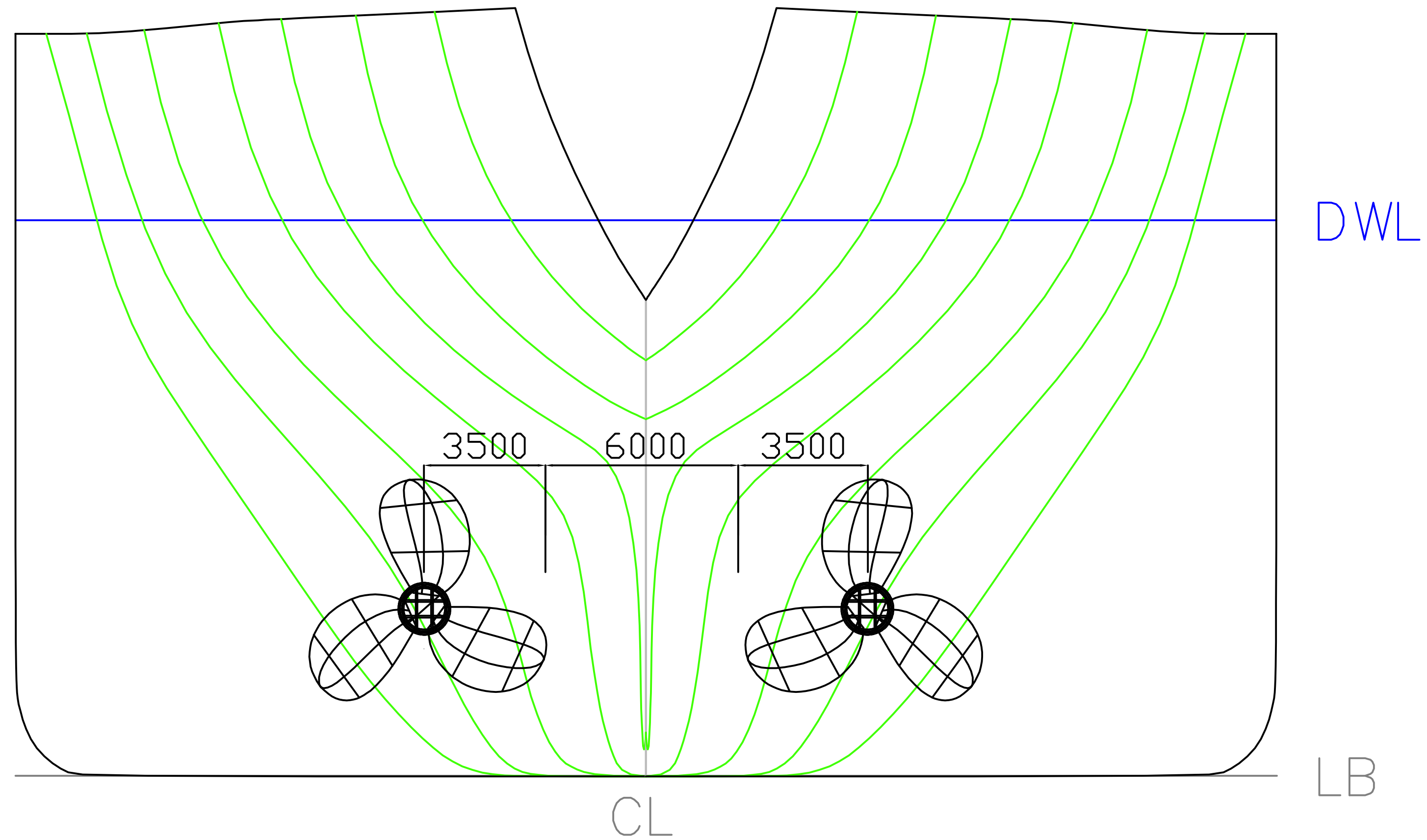
$$RHmax = 15LPP = 220 * 15 = 3.300\ m$$

Por lo que los cumple con la normativa de la IMO.

Capítulo 10. ANEXOS

10.1. ANEXO I PLANO LÍNEA DE EJES





ESCUELA POLITÉCNICA SUPERIOR (UDC)						
ANTEPROYECTO PETROLERO AFRAMAX DE 80.000 T.P.M						
Revisado por: Jose Antonio González Llorente	Revisado por: Marcos M. G.	Aprobado por: Marcos M. G.	Archivo CAD: DG.DWG	Fecha 19/04/2017	Escala 1/200	T. Papel A/3
Máster en Ing. Naval y Oceanica			Descripción del plano: Línea de ejes			
Trabajo fin de máster			Número de proyecto: 17-27		Edición 0	Hoja nº: 1/1

10.2. ANEXO II DOCUMENTACIÓN MOTOR PROPULSOR

12.2 Systems

The system selection comprises a set of alternatives to meet versatile needs. Throughout the whole power range, a simple system and a more sophisticated system are available. A simple system is always more cost-efficient, while a more sophisticated system has other advantages.

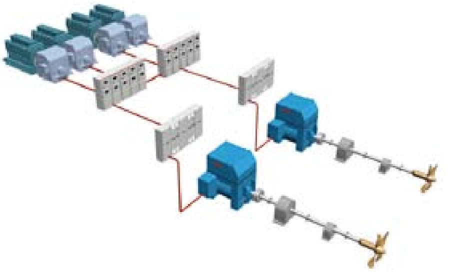
There are six systems available. The fundamental idea was to recognize the versatile needs of various projects and offer distinctive systems that satisfy those needs. Within these six systems, we can offer ready-made solutions that can be ranked and compared according to performance, efficiency, footprint, volume, weight or price – whichever you value the most.

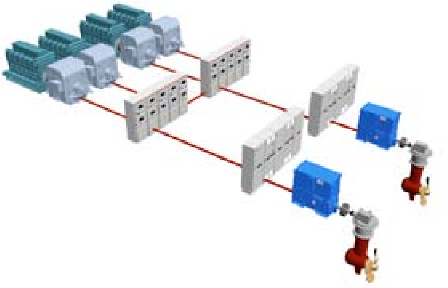
	Single Drive	Single Drive (AFE)	Single Drive (with transformer)	Tandem Drive	Single Drive (twin in/out)	Full Redundant Drive
Direct Drive	X	X	X	X	X	X
Direct Drive Permanent Magnet	X	X	X	X		
High Speed Drive	X	X	X			

Single Drive is the simplest and most cost-efficient system. It comprises a motor, frequency converter, propulsion control and harmonic filter. Single Drives up to 5 MW are low voltage versions. Induction and synchronous motor types are used.

Single Drive (AFE) is equipped with active-front-end frequency converters that can regenerate into the network. Thanks to the regeneration, a separate braking resistor is not needed. Single Drive (AFE) is available for induction motors with up to 5 MW of shaft power. Other parts of the system are as in Single Drive.

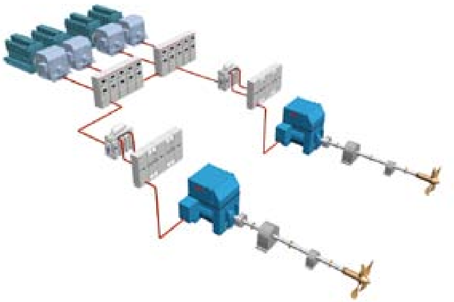
Single Drive and Single Drive (AFE) solutions

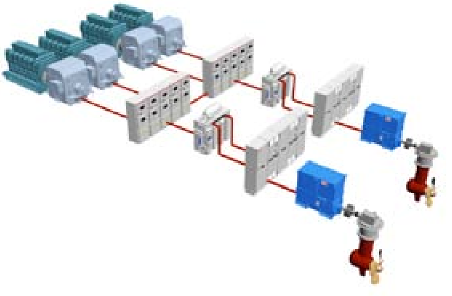




Single Drive (with Transformer) is the first one of systems that incorporate a transformer. Thanks to the transformer, this system has the widest coverage, from 0.8 MW up to 11.5 MW. The system can be integrated into various power plant voltages. Furthermore, the transformer eliminates the need for harmonic filters. Induction and synchronous motor types are used.

Single Drive (with Transformer) solutions

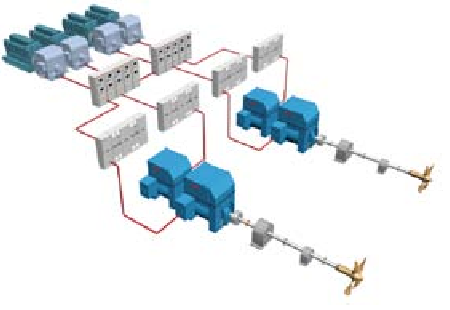




12

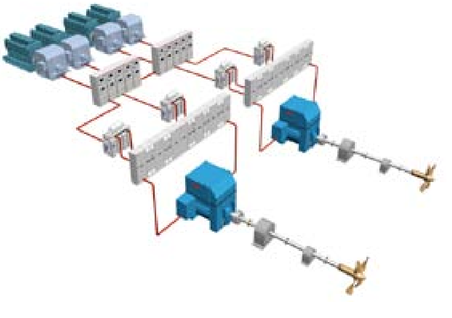
Tandem Drive is a retro design making its way back to the centre of attention. Tandem induction motors rotate a single shaft and provide essential redundancy for the most demanding operations. Two frequency converters will guarantee that a failure will not render the shaft inoperative. The tandem configuration allows the deck height to be lowered, which brings flexibility to projects with critical height restrictions. Tandem Drive is a low voltage solution and can be operated without a certified electrician onboard. Tandem Drives are available from 3.0 MW to 9.6 MW.

Tandem Drive



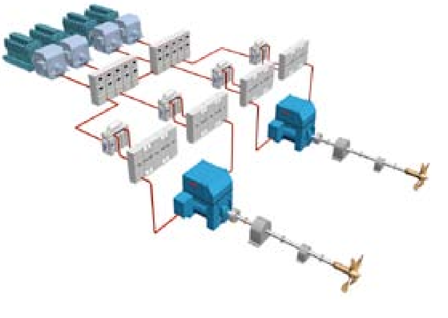
Single Drive (twin in/out) is a twin channel solution for high powers; the frequency converter has a twin supply and it is installed in one lineup. The synchronous motor has two stators. Each channel provides 50% of the propulsion energy. Single Drive (twin in/out) has two transformers that both deliver 50% of the power. Two transformers can be more easily accommodated onboard a vessel instead of one large transformer.

Single Drive (twin in/out) solutions



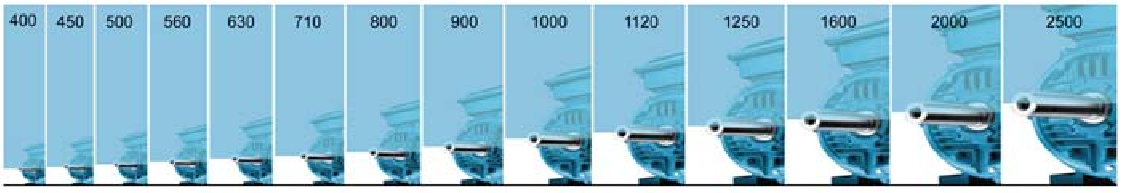
Full Redundant Drive is the most sophisticated system providing redundancy for high powers. Redundancy is an insurance against failures; there is enough redundancy to keep the propeller rotating even in the worst case scenario. The system has a synchronous motor with two stators, two frequency converters, two transformers and two excitation transformers. Shaft powers up to 20 MW can be achieved.

Full Redundant solutions

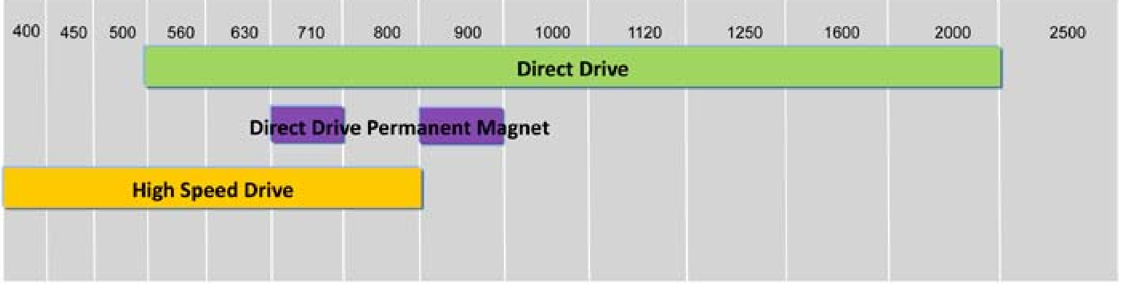


12.3 Sizes and drive steps

ABB propulsion motor frame sizes 400 – 2500 (approx. Equal to shaft height)



Availability of Direct Drive platforms



Sizes
The Direct Drive size refers to the shaft height of the motor, for example, 560 means 560 mm shaft height.

Drive steps
Within sizes, drive steps reflect changes in power that take place when the propeller speed increases. The different drive steps are:

- S
- M
- L
- X
- Y
- Z

12.4 Main components

All main components are designed to withstand extreme electrical and mechanical stress caused by marine operation. Compatibility is ensured by ABB engineering. The motor, frequency converter and propulsion control are always included in the scope of the delivery. The braking resistor, transformer, excitation transformer and harmonic filters are case dependent. The Remote Control System and Uninterrupted Power Supply are optional. The number of the main components depends on the selected system.

The motor meets the requirements specified in IEC 60034 and IEC 60092. The motor is fully enclosed and contains an air-to-water cooling unit. Sufficient cooling is ensured in all load and speed conditions.

The frequency converter utilizes a DTC (Direct Torque Control) method, which guarantees minimum torque ripple and results in the minimum wear of the machinery. The diode front end is a standard rectifier, but an active front end is also available as a system of its own.

The propulsion control is an integral part of the frequency converter and it ensures a safe voyage in all operation conditions. Drive Control Unit (DCU) is a coordinated control system with propulsion functionality and protection. Propulsion Control Unit (PCU) is more a sophisticated and independent control system that allows more customization.

The braking resistor is used to absorb regenerative power.

The transformer is needed to transform the main voltage to appropriate levels for the frequency converter. Transformers contain an air-to-water cooling unit due to high power ratings.

The excitation transformer is utilized with synchronous motors. Excitation transformers are air-cooled due to low power ratings.

Harmonic filters are sometimes needed to filter harmonic distortion in the network.

The Remote Control System (RCS) comprises a human-to-machine interface for controlling the propulsion from the bridge.

The Uninterrupted Power Supply (UPS) keeps the propulsion control systems running in case of interruptions in the mains connection.

12.5 Selection process

With these four simple selections you will find out what is the most cost-effective propulsion solution for your needs.

The selection process is straightforward and consists of four steps after the **propeller speed** and **power** are either known or estimated:

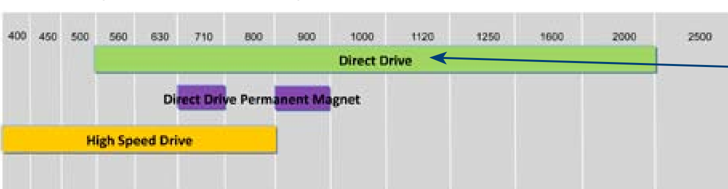
1. Select platform → **2. Select size** → **3. Select system** → **4. Select drive steps**

The list of the pre-calculated drive steps is not exhaustive and they can always be further optimized, but as the comparison of the drive steps shows, the changes that can be achieved are minor. In some cases, however, it might be reasonable to select the next Direct Drive size to achieve gains in efficiency.

Example:
Tank tests indicate that the power of 2000 kW at 220 rpm must be achieved for a shaft. The customer wants to have the most cost-efficient solution.

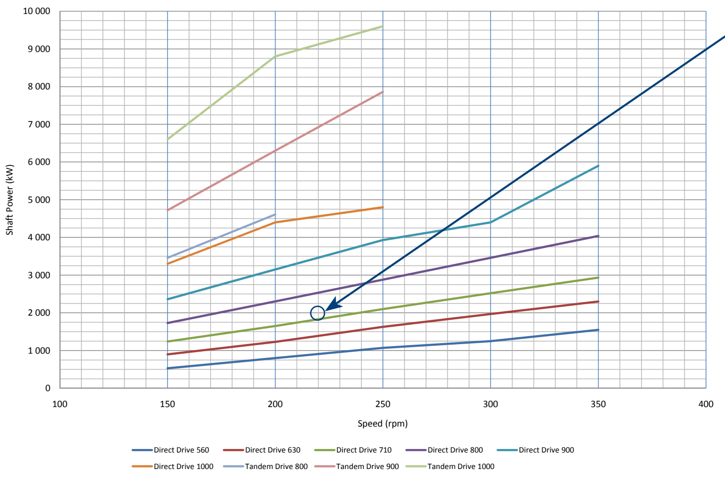
1. Select platform: The customer has compared different propulsion options and operational profiles and decided that the most appropriate and cost-efficient platform is Direct Drive.

Availability of Direct Drive systems



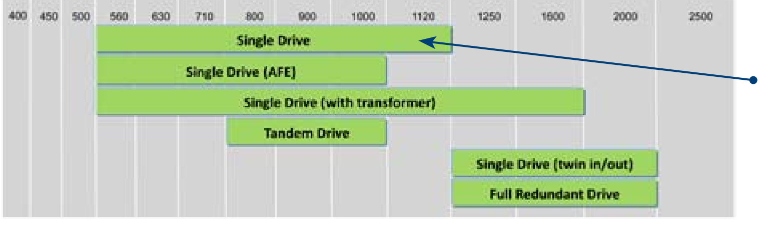
Direct Drive is the most cost-efficient platform over Direct Drive Permanent Magnet. High Speed Drive is not applicable.

2. Select size: The target is 2000 kW at 220 rpm. The performance limit curve indicates Direct Drive 800 for the speed of 220 rpm.



220 rpm requires Direct Drive 800

3. Select system:
Availability of Direct Drive systems



Single Drive is the most cost-efficient system for the example

4. Select drive steps:
Direct Drive 800 M can produce 2305 kW above 200 rpm (page 181), which makes it the proper drive step. The values for Direct Drive 800 M – Single Drive apply (page 181).

Direct Drive 800 – Single Drive					
Drive Step	S	M	L	X	Y
Propeller Speed (rpm)	≥150	≥200	≥250	≥300	≥350
Maximum Power (kW)	1730	2305	2880	3460	4040
Maximum Torque (kNm)	109,7	110,0	109,9	110,4	110,4
Drive (kVA)	2680	3330	3970	4630	5960
Transformer (kVA)	-	-	-	-	-
Braking Capacity (MJ)	21,6	21,6	21,6	21,6	21,6

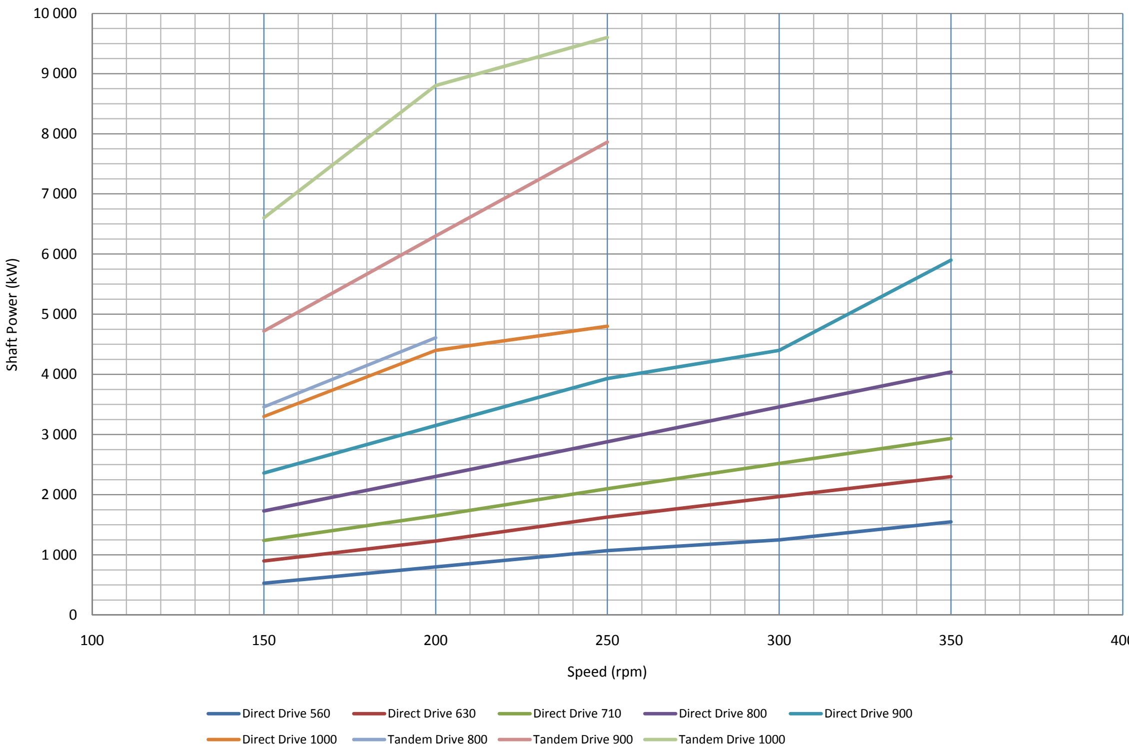
DESIGNER TIP:

Read more about the System selection (pages 166 – 168)

13 Technical specification for Direct Drive

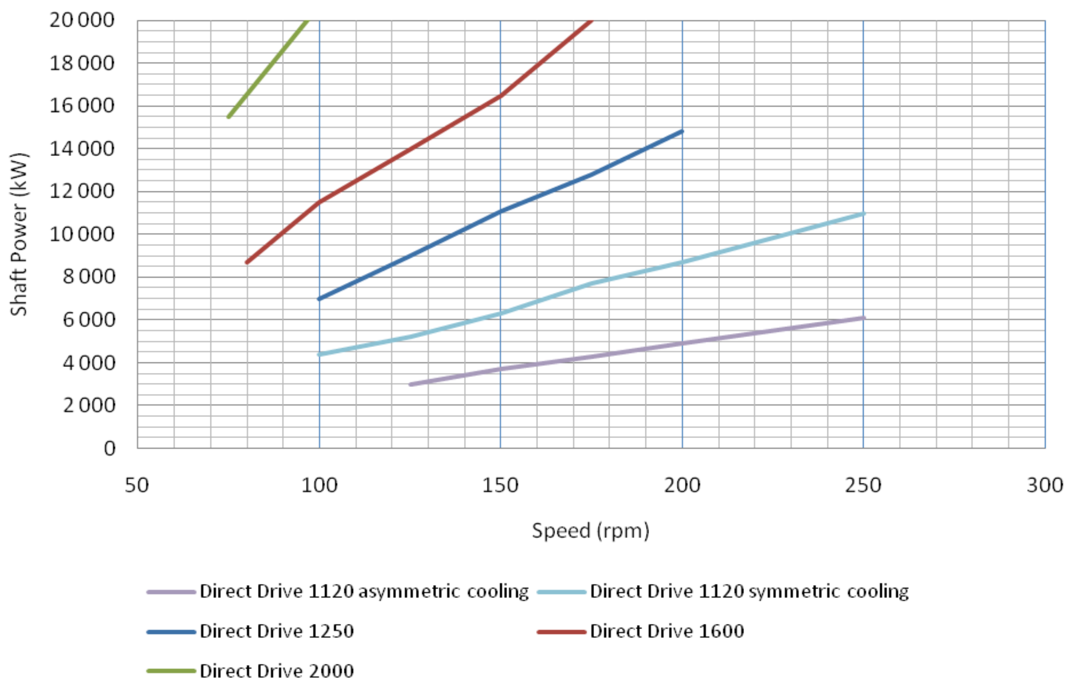
The performance range for Direct Drive is divided into two graphs. The first graph charts the performance ranges for the sizes 560 to 1000. The second graph charts sizes 1120 to 2000. An appropriate drive size can be found in the graphs below.

Performance limits for Direct Drives 560 – 1000 (Figure 1)



13

Performance limits for Direct Drives 1120 – 2000. (Figure 2)



Power range (MW) for each system (check performance limits from figures 1 and 2)

	Single Drive	Single Drive (AFE)	Single Drive (with Trans-former)	Tandem Drive	Single Drive (twin in/out)	Full Redundant Drive
Power Range	0.5…6.3	0.5…5.1	0.8…11.5	3.4…9.6	12.8…20.0	11.1…20.0

Availability of Direct Drive systems

The table below summarizes the main components in each system.

Number of main components – Direct Drive						
	Single Drive	Single Drive (AFE)	Single Drive (with Trans-former)	Tandem Drive	Single Drive (twin in/out)	Full Redundant Drive
Motor	1	1	1	2	1	1
Frequency converter	1	1	1	2	1	2
Braking resistor	1	0	1	2	1	2
Transformer	0	0	1	0	2	2
Excitation transformer	0, 1)	0	0, 1)	0	1	2
Harmonic filter	1	0	0	1	0	0

1) One excitation transformer in case of a synchronous motor

System features – Direct Drive						
	Single Drive	Single Drive (AFE)	Single Drive (with Trans-former)	Tandem Drive	Single Drive (twin in/out)	Full Redundant Drive
Motor	Induction up to size 1000, synchronous from size 1120	Induction	Induction up to size 1000, synchronous from size 1120	Induction	Synchronous	Synchronous
Stator system(s) in a motor	1	1	1, 1)	1	2	2
Frequency converter	LV ≤ ~5 MW MV ≥ ~5 MW	LV	LV ≤ ~5 MW MV ≥ ~5 MW	LV	MV	MV
Pulse number	6	AFE	12	6	24	24
Propulsion control	Drive Control Unit (DCU), Propulsion Control Unit (PCU) as an option					PCU
Remote Control System	Interface towards Remote Control System, RCS as an option					
Uninterrupted Power Supply	UPS as an option					

1) After 10 MW two stator systems

13

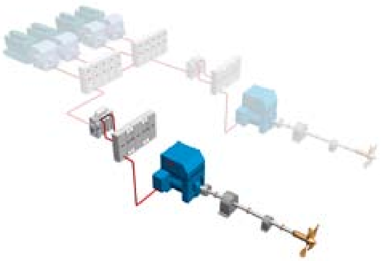
Technical data for the main components

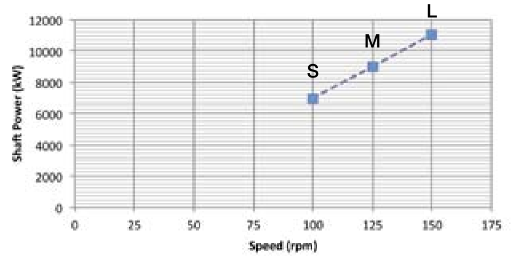
Technical data – Direct Drive						
	Motor	Frequency converter	Braking resistor	Transformer	Excitation transformer	Harmonic filter
Max. ambient temperature (°C)	50	45	45	45	45	45
Max. cooling water tempera-ture (°C)	36	36	N/A	36	N/A	36
Method of cooling	IC86W	Direct Liquid Cooling	Air cooled	AFWF	AN	Air cooled
Installation	IM1001 / IM7315, [1]	One Lineup	Wall standing	Feet for welding	Feet for welding	Wall standing
Enclosure	IP44	IP42 / IP32 [2]	IP23	IP44	IP23	IP23
Color	MUNSELL8B	RAL7035	RAL7035	RAL7035	RAL7035	RAL7035
Insulation class [3]	F	N/A	N/A	F	F	N/A
Temperature rise class [3]	F	N/A	N/A	F	F	N/A
Cabling direction	Bottom	Bottom	Bottom	Bottom	Bottom	Bottom
Piping direction	Side	Side	N/A	Side	N/A	N/A

[1] IM7315 for Direct Drives sizes 1250 and above
[2] Low voltage converter IP42, Medium voltage converter IP32
[3] According to DNV

REMARK:

Direct Drive is a pre-engineered solution and to guarantee a successful delivery, ABB’s installation, cabling and operation instructions must be followed.



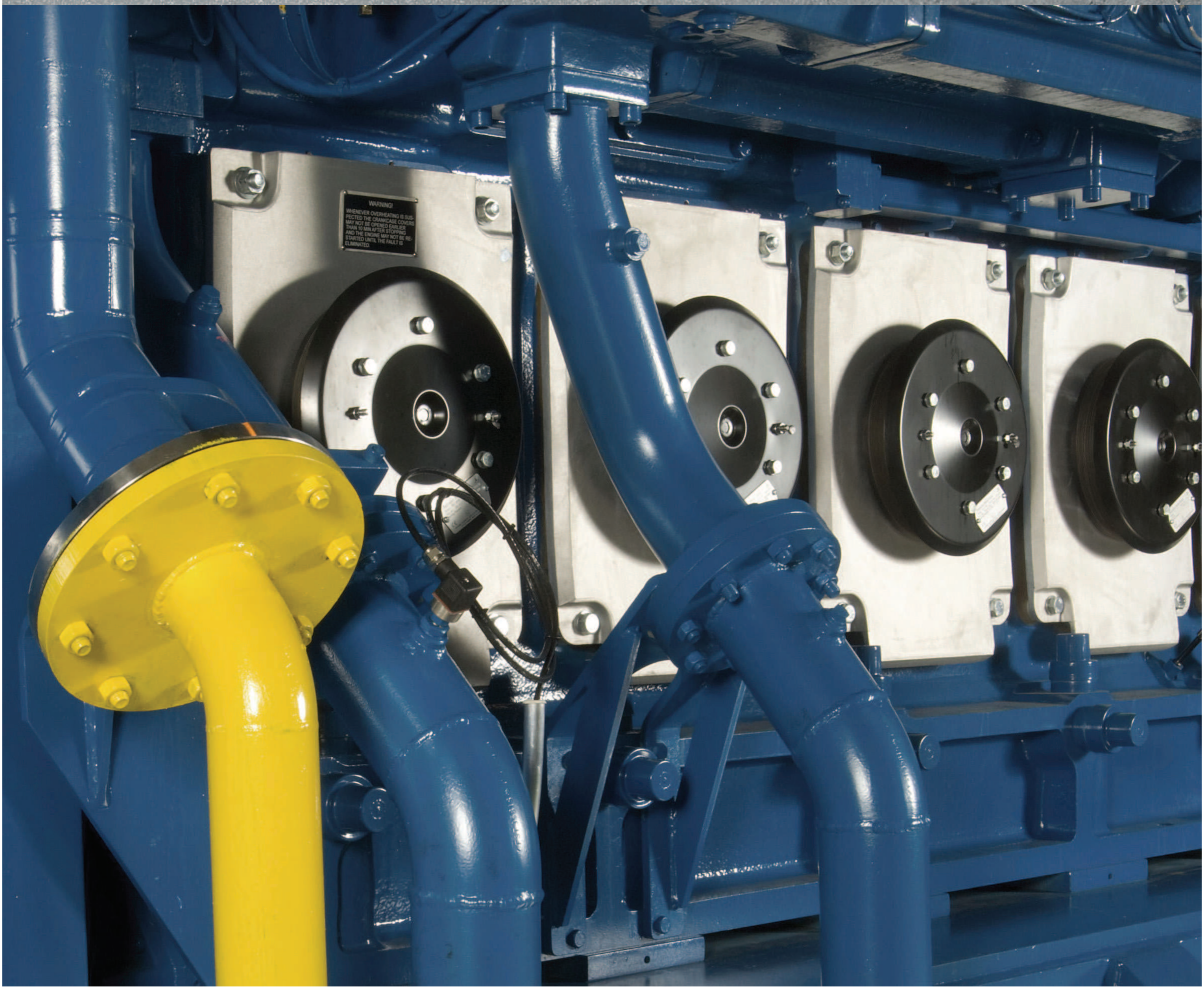


	Direct Drive 1250 - Single Drive (with Transformer)					
	Drive Step	S	M	L	X	Y
	Propeller Speed (rpm)	≥100	≥125	≥150	-	-
	Maximum Power (kW)	7000	9000	11100	-	-
	Maximum Torque (kNm)	669,0	687,5	706,6	-	-
	Drive (kVA)	9000	11000	14000	-	-
	Transformer (kVA)	8000	12000	14000	-	-
	Braking Capacity (MJ)	30	46	46	-	-
Drivetrain Efficiency (%)	Motor	94,69	95,6	96,53	-	-
	Frequency Converter	98,5	98,5	98,5	-	-
	Transformer	99	99	99	-	-
	Total Electrical Efficiency	92,3	93,2	94,1	-	-
Main Connection	Input Voltage (VAC)	6600 / 11000	6600 / 11000	6600 / 11000	-	-
	Frequency (Hz)	50/60	50/60	50/60	-	-
	Power factor	0,95	0,95	0,95	-	-
	Input power (kVA)	7980	10162	12413	-	-
	Input Current (A)	699 / 419	889 / 534	1086 / 652	-	-
Footprint (m^2)	Motor	24,4	24,4	24,4	-	-
	Frequency Converter	9,3	9,8	12,7	-	-
	Braking Resistor	1,6	2,0	2,0	-	-
	Transformer	10,9	12,5	13,3	-	-
	Excitation Transformer	0,8	0,8	1,5	-	-
	Harmonic Filter	-	-	-	-	-
	Total	47,0	49,5	53,8	-	-
Dimensions (L x W xH)	Motor	5940 x 4100 x 3925	5940 x 4100 x 3925	5940 x 4100 x 3925	-	-
	Frequency Converter	7930 x 1176 x 2475	8330 x 1176 x 2475	10830 x 1176 x 2475	-	-
	Braking Resistor	1800 x 900 x 1700	2200 x 900 x 1700	2200 x 900 x 1700	-	-
	Transformer	4200 x 2600 x 2950	5000 x 2500 x 3300	5300 x 2500 x 3500	-	-
	Excitation Transformer	1230 x 670 x 1355	1230 x 670 x 1355	1240 x 1170 x 1555	-	-
	Harmonic Filter	-	-	-	-	-
Weight (kg)	Motor	84050	84100	84100	-	-
	Frequency Converter	6600	6800	8900	-	-
	Braking Resistor	620	750	750	-	-
	Transformer	12500	17200	18500	-	-
	Excitation Transformer	1330	1330	1720	-	-
	Harmonic Filter	-	-	-	-	-
	Total	103770	108850	112250	-	-
LT-water flow (m^3/h)	Motor	70	73	71	-	-
	Frequency Converter	12,1	15,0	17,6	-	-
	Braking Resistor	-	-	-	-	-
	Transformer	19	19	19	-	-
	Excitation Transformer	-	-	-	-	-
	Harmonic Filter	-	-	-	-	-
Losses to water (kW)	Motor	373	394	379	-	-
	Frequency Converter	101,1	124,7	146,8	-	-
	Braking Resistor	-	-	-	-	-
	Transformer	158	158	158	-	-
	Excitation Transformer	-	-	-	-	-
	Harmonic Filter	-	-	-	-	-
	Total	474	518	526	-	-
Losses to ambient (kW)	Motor	20	21	20	-	-
	Frequency Converter	12,3	14,1	15,8	-	-
	Braking Resistor	Intermittent	Intermittent	Intermittent	-	-
	Transformer	9,2	11,2	12,2	-	-
	Excitation Transformer	5,4	5,4	7	-	-
	Harmonic Filter	-	-	-	-	-
	Total	46,5	51,4	55,0	-	-

WÄRTSILÄ

WÄRTSILÄ Engines

WÄRTSILÄ 34DF PRODUCT GUIDE



1. Main Data and Outputs

1.1 Technical main data

The Wärtsilä 34DF is a 4-stroke, non-reversible, turbocharged and inter-cooled dual fuel engine with direct injection of liquid fuel and indirect injection of gas fuel. The engine can be operated in gas mode or in diesel mode.

Cylinder bore	340 mm
Stroke	400 mm
Piston displacement	36.3 l/cyl
Number of valves	2 inlet valves and 2 exhaust valves
Cylinder configuration	6, 8 and 9 in-line; 12 and 16 in V-form
Direction of rotation	clockwise, counterclockwise on request
Speed	720, 750 rpm
Mean piston speed	9.6, 10.0 m/s

1.2 Maximum continuous output

Table 1-1 Rating table for Wärtsilä 34DF

Cylinder configuration	Main engines 750 rpm	Generating sets			
		720 rpm		750 rpm	
	Engine [kW]	Engine [kW]	Generator [kVA]	Engine [kW]	Generator [kVA]
Wärtsilä 6L34DF	3000	2880	3460	3000	3600
Wärtsilä 8L34DF	4000	3840	4610	4000	4800
Wärtsilä 9L34DF	4500	4320	5180	4500	5400
Wärtsilä 12V34DF	6000	5760	6910	6000	7200
Wärtsilä 16V34DF	8000	7680	9220	8000	9600

The mean effective pressure P_e can be calculated using the following formula:

$$P_e = \frac{P \times c \times 1.2 \times 10^9}{D^2 \times L \times n \times \pi}$$

where:

P_e = mean effective pressure [bar]

P = output per cylinder [kW]

n = engine speed [r/min]

D = cylinder diameter [mm]

L = length of piston stroke [mm]

c = operating cycle (4)

1.6.2 Generating sets

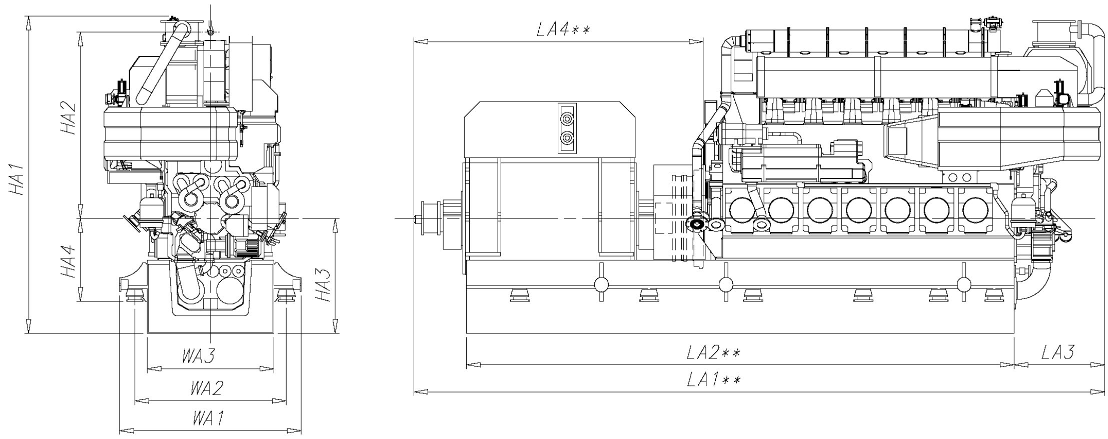


Fig 1-5 In-line engines (DAAE082427)

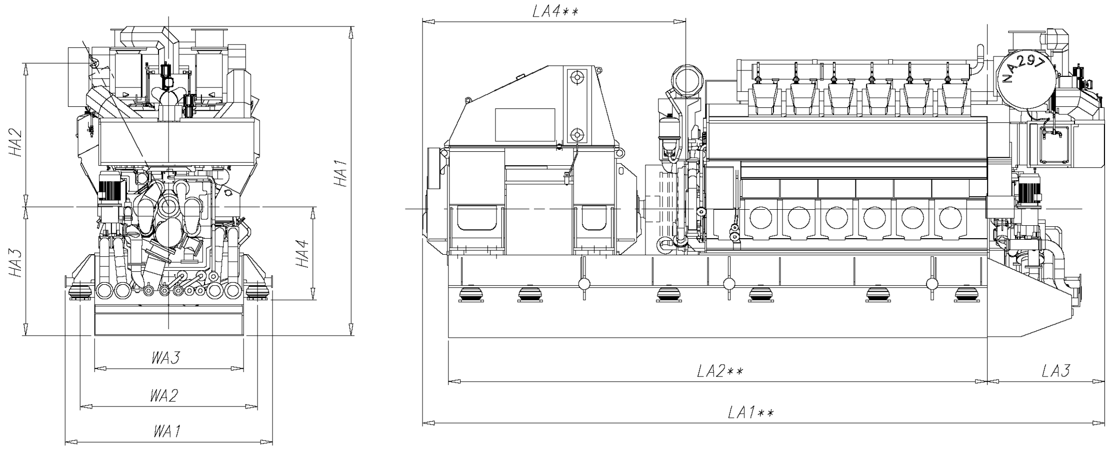


Fig 1-6 V engines (DAAE082975)

Engine	kW/cyl	LA1**	LA2**	LA3	LA4**	WA1	WA2	WA3	HA1	HA2	HA3	HA4	Weight**
W 6L34DF	480	8765	6900	1215	3160	2290	1910	1600	4000	2345	1450	1055	60
Wärtsilä 8L34DF	480	10410	8650	1285	3645	2690	2310	2000	4180	2345	1630	1055	76
W 9L34DF	480	10610	8850	1285	3845	2890	2510	2200	4180	2345	1630	1055	87
W 12V34DF	480	10260	7950	1985	3775	3060	2620	2200	4335	2120	1900	1375	99
W 16V34DF	480	11465	9130	1925	3765	3360	2920	2500	4445	2120	1850	1375	124

** Dependent on generator and flexible coupling.

All dimensions in mm. Weight in metric tons with liquids.

2. Operating Ranges

2.1 Engine operating range

Below nominal speed the load must be limited according to the diagrams in this chapter in order to maintain engine operating parameters within acceptable limits. Operation in the shaded area is permitted only temporarily during transients. Minimum speed is indicated in the diagram, but project specific limitations may apply.

2.1.1 Controllable pitch propellers

An automatic load control system is required to protect the engine from overload. The load control reduces the propeller pitch automatically, when a pre-programmed load versus speed curve ("engine limit curve") is exceeded, overriding the combinator curve if necessary. Engine load is determined from measured shaft power and actual engine speed. The shaft power meter is Wärtsilä supply.

The propulsion control must also include automatic limitation of the load increase rate. Maximum loading rates can be found later in this chapter.

The propeller efficiency is highest at design pitch. It is common practice to dimension the propeller so that the specified ship speed is attained with design pitch, nominal engine speed and 85% output in the specified loading condition. The power demand from a possible shaft generator or PTO must be taken into account. The 15% margin is a provision for weather conditions and fouling of hull and propeller. An additional engine margin can be applied for most economical operation of the engine, or to have reserve power.

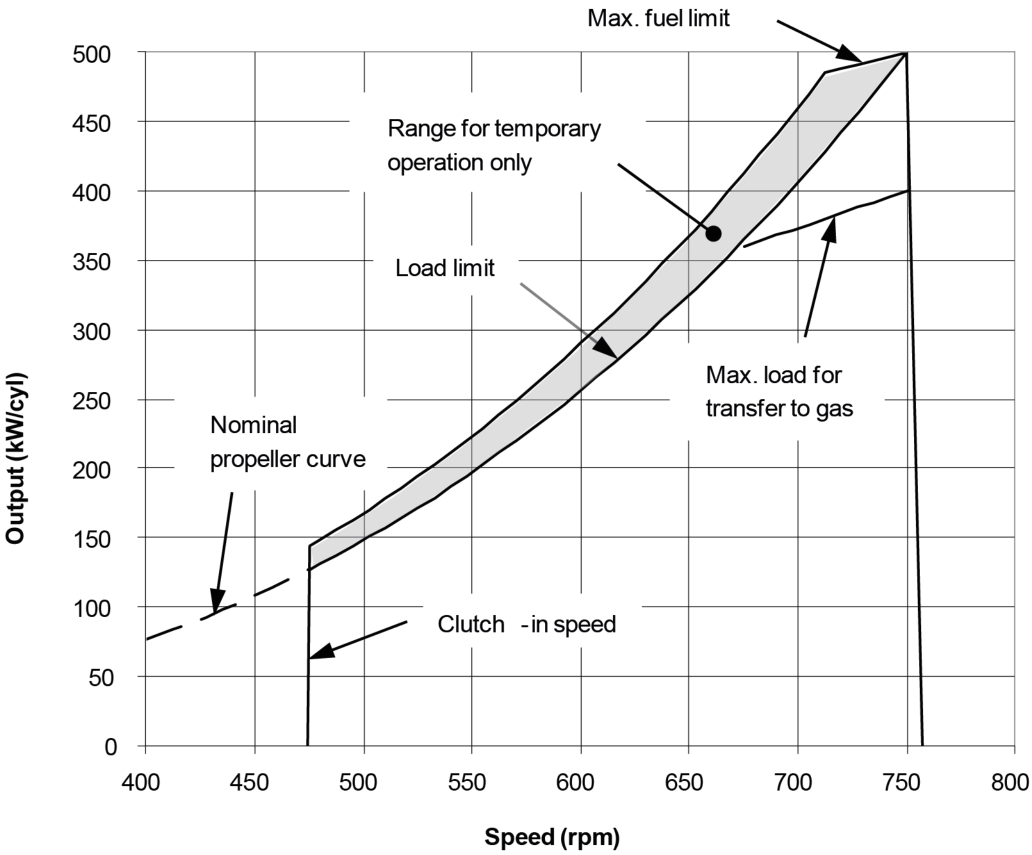


Fig 2-1 **Operating field for CP Propeller, rated speed 750 rpm**

Remarks: The maximum output may have to be reduced depending on gas properties and gas pressure, refer to section "*Derating of output in gas mode*". The permissible output will in such case be reduced with same percentage at all revolution speeds.

Restrictions for low load operation to be observed.

2.2 Loading capacity

Controlled load increase is essential for highly supercharged engines, because the turbocharger needs time to accelerate before it can deliver the required amount of air. Sufficient time to achieve even temperature distribution in engine components must also be ensured. Dual fuel engines operating in gas mode require precise control of the air/fuel ratio, which makes controlled load increase absolutely decisive for proper operation on gas fuel.

The loading ramp “preheated, normal gas” (see figures) can be used as the default loading rate for both diesel and gas mode. If the control system has only one load increase ramp, then the ramp for a preheated engine must be used. The HT-water temperature in a preheated engine must be at least 60°C, preferably 70°C, and the lubricating oil temperature must be at least 40°C.

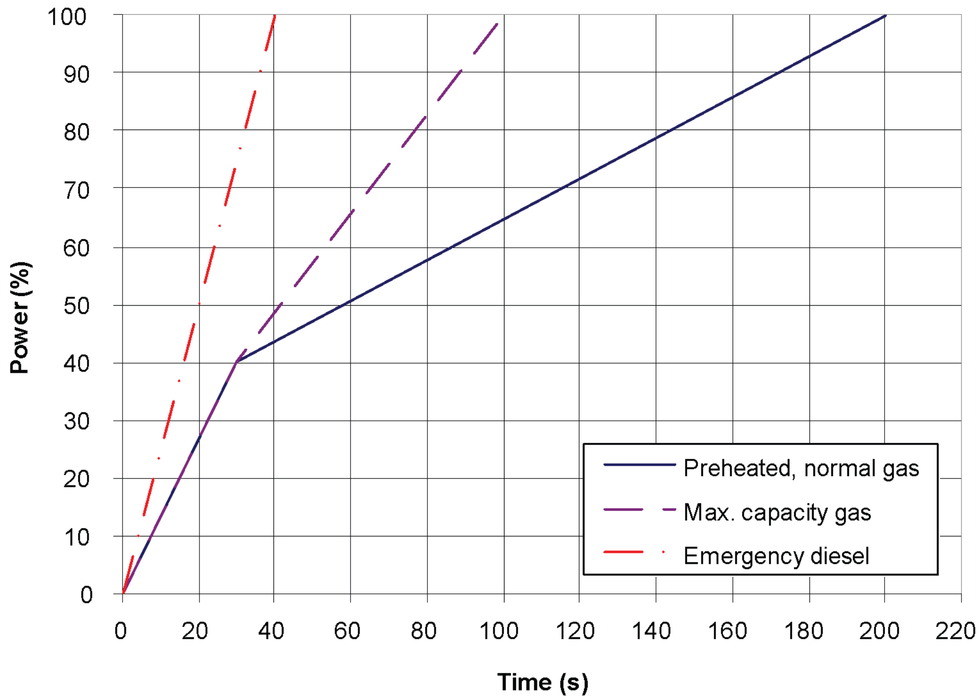
The loading ramp “max. capacity gas” indicates the maximum capability of the engine in gas mode. Faster loading may result in alarms, knock and undesired trips to diesel. This ramp can also be used as normal loading rate in diesel mode once the engine has attained normal operating temperature.

The maximum loading rate “emergency diesel” is close to the maximum capability of the engine in diesel mode. It shall not be used as the normal loading rate in diesel mode.

The load should always be applied gradually in normal operation. Acceptable load increments are smaller in gas mode than in diesel mode and also smaller at high load, which must be taken into account in applications with sudden load changes. The time between load increments must be such that the maximum loading rate is not exceeded. In the case of electric power generation, the classification society shall be contacted at an early stage in the project regarding system specifications and engine loading capacity.

Electric generators must be capable of 10% overload. The maximum engine output is 110% in diesel mode and 100% in gas mode. Transfer to diesel mode takes place automatically in case of overload. Lower than specified methane number may also result in automatic transfer to diesel when operating close to 100% output. Expected variations in gas fuel quality and momentary load level must be taken into account to ensure that gas operation can be maintained in normal operation.

2.2.1 Mechanical propulsion, controllable pitch propeller (CPP)



* For engines operating at nominal speed, see loading ramps for DE propulsion.

Fig 2-2 Maximum load increase rates for variable speed engines

The propulsion control must not permit faster load reduction than 20 s from 100% to 0% without automatic transfer to diesel first.

2.2.2 Constant speed applications

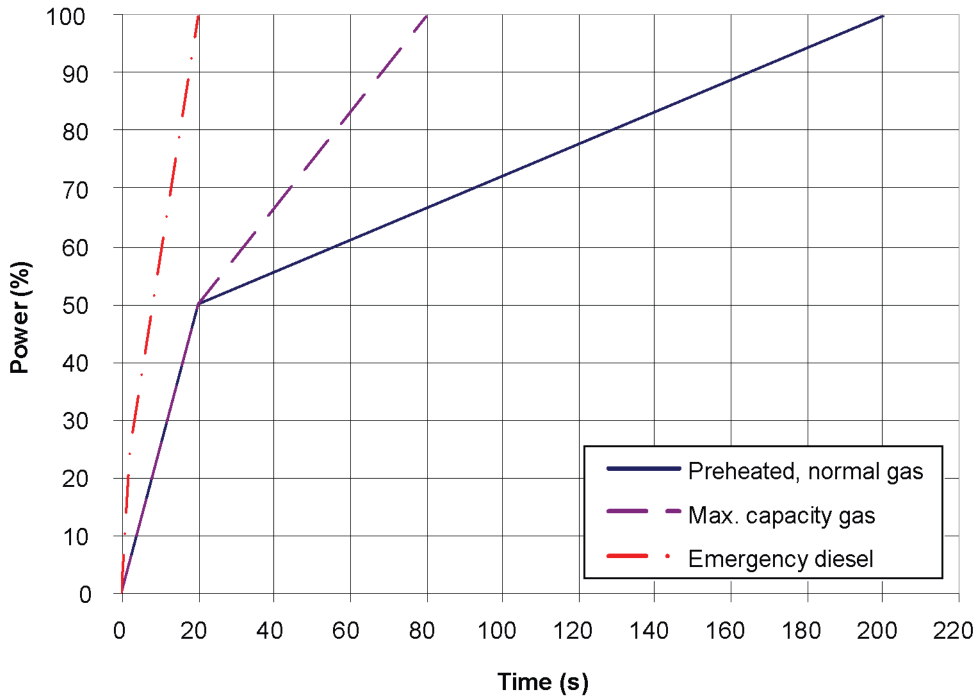


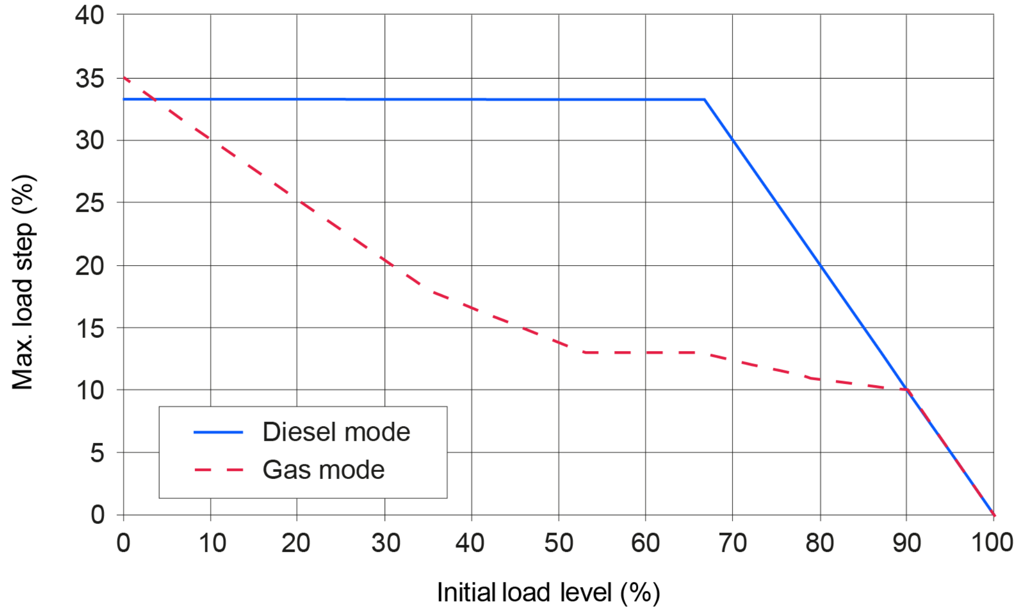
Fig 2-3 Increasing load successively from 0 to 100% MCR

The propulsion control and the power management system must not permit faster load reduction than 20 s from 100% to 0% without automatic transfer to diesel first.

In electric propulsion applications loading ramps are implemented both in the propulsion control and in the power management system, or in the engine speed control in case isochronous load sharing is applied. When the load sharing is based on speed droop, it must be taken into account that the load increase rate of a recently connected generator is the sum of the load transfer performed by the power management system and the load increase performed by the propulsion control.

2.2.2.1 Maximum instant load steps

The electrical system must be designed so that tripping of breakers can be safely handled. This requires that the engines are protected from load steps exceeding their maximum load acceptance capability. If fast load shedding is complicated to implement or undesired, the instant load step capacity can be increased with a fast acting signal that requests transfer to diesel mode.



When performing the electric load analysis for the vessel in various operating conditions, evaluate possible scenarios that cause sudden load changes and check against the engine capacity in gas mode as shown in the diagram.

Fig 2-4 Maximum instant load steps in % of MCR, 500 kW/cyl

2.2.2.1.1 Gas mode

- Maximum step-wise load increases according to figure
- Steady-state frequency band ≤ 1.5 %
- Maximum speed drop 10 %
- Recovery time ≤ 10 s
- Time between load steps of maximum size ≥ 15 s
- Maximum step-wise load reductions: 100-75-45-0%

2.2.2.1.2 Diesel mode

- Maximum step-wise load increase 33% of MCR
- Steady-state frequency band ≤ 1.0 %

- Maximum speed drop 10 %
- Recovery time ≤ 5 s
- Time between load steps of maximum size ≥ 8 s

2.2.2.1.3 Start-up

A stand-by generator reaches nominal speed in 50-70 seconds after the start signal (check of pilot fuel injection is always performed during a normal start).

With black-out start active nominal speed is reached in about 25 s (pilot fuel injection disabled).

The engine can be started with gas mode selected provided that the engine is preheated and the air receiver temperature is at required level. It will then start on MDF and gas fuel will be used as soon as the pilot check is completed and the gas supply system is ready.

Start and stop on heavy fuel is not restricted.

2.3 Operation at low load and idling

Absolute idling (declutched main engine, disconnected generator):

- Maximum 10 minutes if the engine is to be stopped after the idling. 3-5 minutes idling before stop is recommended.
- Maximum 8 hours if the engine is to be loaded after the idling.

Operation below 20 % load on HFO or below 10 % load on MDF or gas

- Maximum 100 hours continuous operation. At intervals of 100 operating hours the engine must be loaded to minimum 70 % of the rated output for 1 hour. Before operating below 10% in gas mode, the engine must run above 10% load for at least 10 minutes. It is however acceptable to change to gas mode directly after the engine has reached nominal speed after the engine has started, provided that the charge air temperature is 55 °C.

Operation above 20 % load on HFO or above 10 % load on MDF or gas

- No restrictions.

2.4 Low air temperature

In cold conditions the following minimum inlet air temperatures apply:

Gas mode:

- Low load + 5ºC
- High load -10ºC

Diesel mode:

- Starting + 5ºC
- Idling - 5ºC
- High load - 10ºC

The two-stage charge air cooler is useful for heating of the charge air during prolonged low load operation in cold conditions. Sustained operation between 0 and 40% load can however require special provisions in cold conditions to prevent too low HT-water temperature. If necessary, the preheating arrangement can be designed to heat the running engine (capacity to be checked).

For further guidelines, see chapter *Combustion air system design*.

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3. Technical Data

3.1 Wärtsilä 6L34DF

Wärtsilä 6L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Engine speed	rpm	720		750		720		750		750		750	
Engine output	kW	2880		3000		2880		3000		3000		3000	
Mean effective pressure	MPa	2.2		2.2		2.2		2.2		2.2		2.2	
Speed mode		Constant		Constant		Constant		Constant		Constant		Variable	
IMO compliance		Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2
Combustion air system (Note 1)													
Flow at 100% load	kg/s	4.5	5.4	4.5	5.4	4.5	5.4	4.5	5.4	4.5	5.4	4.5	5.5
Temperature at turbocharger intake, max.	°C	45		45		45		45		45		45	
Temperature after air cooler (TE 601), load > 70%	°C	45	-	45	-	45	-	45	-	45	-	45	-
Temperature after air cooler (TE 601), load 30...70%	°C	55	-	55	-	55	-	55	-	55	-	55	-
Temperature after air cooler (TE 601)	°C	-	50	-	50	-	50	-	50	-	50	-	50
Exhaust gas system (Note 2)													
Flow at 100% load	kg/s	4.6	5.5	4.6	5.5	4.6	5.5	4.6	5.5	4.6	5.5	4.6	5.6
Flow at 75% load	kg/s	3.8	4.4	3.8	4.4	3.8	4.4	3.8	4.4	3.8	4.4	3.7	4.3
Flow at 50% load	kg/s	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.0	3.1
Temperature after turbocharger at 100% load (TE 517)	°C	362	355	381	381	362	346	381	370	381	370	381	361
Temperature after turbocharger at 75% load (TE 517)	°C	383	327	401	349	383	318	401	340	401	340	386	348
Temperature after turbocharger at 50% load (TE 517)	°C	386	350	402	371	386	346	402	366	402	366	340	333
Backpressure, max.	kPa	4		4		4		4		4		4	
Calculated exhaust diameter for 35 m/s	mm	545	596	553	608	545	591	553	603	553	603	553	605
Heat balance at 100% load (Note 3)													
Jacket water, HT-circuit	kW	357	410	372	430	357	406	372	425	372	425	372	443
Charge air, HT-circuit	kW	601	933	601	933	601	933	601	933	601	933	601	966
Charge air, LT-circuit	kW	171	179	171	179	171	179	171	179	171	179	171	184
Lubricating oil, LT-circuit	kW	250	252	259	264	250	250	260	261	260	261	260	281
Radiation	kW	115	117	120	123	115	116	120	121	120	121	120	123
Fuel consumption (Note 4)													
Total energy consumption at 100% load	kJ/kWh	7470	-	7470	-	7470	-	7470	-	7470	-	7470	-
Total energy consumption at 85% load	kJ/kWh	7620	-	7620	-	7620	-	7620	-	7620	-	7570	-

Wärtsilä 6L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Total energy consumption at 75% load	kJ/kWh	7850	-	7850	-	7850	-	7850	-	7850	-	7590	-
Total energy consumption at 50% load	kJ/kWh	8600	-	8600	-	8600	-	8600	-	8600	-	7790	-
Fuel gas consumption at 100% load	kJ/kWh	7387	-	7387	-	7387	-	7387	-	7387	-	7387	-
Fuel gas consumption at 85% load	kJ/kWh	7527	-	7527	-	7527	-	7527	-	7527	-	7471	-
Fuel gas consumption at 75% load	kJ/kWh	7743	-	7743	-	7743	-	7743	-	7743	-	7478	-
Fuel gas consumption at 50% load	kJ/kWh	8435	-	8435	-	8435	-	8435	-	8435	-	7643	-
Fuel oil consumption at 100% load	g/kWh	1.9	191	1.9	192	1.9	189	1.9	190	1.9	190	1.9	190
Fuel oil consumption at 85% load	g/kWh	2.2	188	2.2	189	2.2	186	2.2	187	2.2	187	2.2	186
Fuel oil consumption at 75% load	g/kWh	2.5	188	2.5	189	2.5	186	2.5	187	2.5	187	2.5	184
Fuel oil consumption 50% load	g/kWh	3.8	194	3.8	195	3.8	194	3.8	195	3.8	195	3.4	183
Fuel gas system (Note 5)													
Gas pressure at engine inlet, min (PT901)	kPa (a)	535	-	535	-	535	-	535	-	535	-	535	-
Gas pressure to Gas Valve Unit, min	kPa (a)	655	-	655	-	655	-	655	-	655	-	655	-
Gas temperature before Gas Valve Unit	°C	0...60	-	0...60	-	0...60	-	0...60	-	0...60	-	0...60	-
Fuel oil system													
Pressure before injection pumps (PT 101)	kPa	700±50		700±50		700±50		700±50		700±50		700±50	
Fuel oil flow to engine, approx	m^3/h	3.1		3.2		3.1		3.2		3.2		3.2	
HFO viscosity before the engine	cSt	-	16...24	-	16...24	-	16...24	-	16...24	-	16...24	-	16...24
Max. HFO temperature before engine (TE 101)	°C	-	140	-	140	-	140	-	140	-	140	-	140
MDF viscosity, min.	cSt	2.0		2.0		2.0		2.0		2.0		2.0	
Max. MDF temperature before engine (TE 101)	°C	45		45		45		45		45		45	
Leak fuel quantity (HFO), clean fuel at 100% load	kg/h		2.2		2.3		2.2		2.3		2.3		2.4
Leak fuel quantity (MDF), clean fuel at 100% load	kg/h	5.6	11.1	5.8	11.6	5.6	11.1	5.8	11.6	5.8	11.6	5.9	11.8
Pilot fuel (MDF) viscosity before the engine	cSt	2...11		2...11		2...11		2...11		2...11		2...11	
Pilot fuel pressure at engine inlet (PT 112)	kPa (a)	550...750		550...750		550...750		550...750		550...750		550...750	
Pilot fuel pressure drop after engine, max	kPa	150		150		150		150		150		150	
Pilot fuel return flow at 100% load	kg/h	590		590		590		590		590		590	
Lubricating oil system													
Pressure before bearings, nom. (PT 201)	kPa	500		500		500		500		500		500	
Suction ability, including pipe loss, max.	kPa	30		30		30		30		30		30	
Priming pressure, nom. (PT 201)	kPa	50		50		50		50		50		50	

Wärtsilä 6L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Suction ability priming pump, including pipe loss, max.	kPa	30		30		30		30		30		30	
Temperature before bearings, nom. (TE 201)	°C	63		63		63		63		63		63	
Temperature after engine, approx.	°C	78		78		78		78		78		78	
Pump capacity (main), engine driven	m³/h	78		81		78		81		81		81	
Pump capacity (main), electrically driven	m³/h	67		70		67		70		70		70	
Priming pump capacity (50/60Hz)	m³/h	15.0 / 18.0		15.0 / 18.0		15.0 / 18.0		15.0 / 18.0		15.0 / 18.0		15.0 / 18.0	
Oil volume, wet sump, nom.	m³	1.6		1.6		1.6		1.6		1.6		1.6	
Oil volume in separate system oil tank	m³	3		3		3		3		3		3	
Oil consumption at 100% load, approx.	g/kWh	0.4		0.4		0.4		0.4		0.4		0.4	
Crankcase ventilation flow rate at full load	l/min	840		840		840		840		840		840	
Crankcase ventilation backpressure, max.	kPa	0.3		0.3		0.3		0.3		0.3		0.3	
Oil volume in turning device	l	...		...		...		...		...		...	
Oil volume in speed governor	l	1.4...2.2		1.4...2.2		1.4...2.2		1.4...2.2		1.4...2.2		1.4...2.2	
HT cooling water system													
Pressure at engine, after pump, nom. (PT 401)	kPa	250 + static		250 + static		250 + static		250 + static		250 + static		250 + static	
Pressure at engine, after pump, max. (PT 401)	kPa	530		530		530		530		530		530	
Temperature before cylinders, approx. (TE 401)	°C	85		85		85		85		85		85	
Temperature after engine, nom.	°C	96		96		96		96		96		96	
Capacity of engine driven pump, nom.	m³/h	60		60		60		60		60		60	
Pressure drop over engine, total	kPa	100		100		100		100		100		100	
Pressure drop in external system, max.	kPa	100		100		100		100		100		100	
Pressure from expansion tank	kPa	70...150		70...150		70...150		70...150		70...150		70...150	
Water volume in engine	m³	0.41		0.41		0.41		0.41		0.41		0.41	
Delivery head of stand-by pump	kPa	250		250		250		250		250		250	
LT cooling water system													
Pressure at engine, after pump, nom. (PT 471)	kPa	250+ static		250+ static		250+ static		250+ static		250+ static		250+ static	
Pressure at engine, after pump, max. (PT 471)	kPa	530		530		530		530		530		530	
Temperature before engine, max. (TE 471)	°C	38		38		38		38		38		38	
Temperature before engine, min. (TE 471)	°C	25		25		25		25		25		25	
Capacity of engine driven pump, nom.	m³/h	60		60		60		60		60		60	
Pressure drop over charge air cooler	kPa	35		35		35		35		35		35	
Pressure drop in external system, max.	kPa	100		100		100		100		100		100	

Wärtsilä 6L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Pressure from expansion tank	kPa	70...150		70...150		70...150		70...150		70...150		70...150	
Delivery head of stand-by pump	kPa	250		250		250		250		250		250	
Starting air system (Note 6)													
Pressure, nom.	kPa	3000		3000		3000		3000		3000		3000	
Pressure, max.	kPa	3000		3000		3000		3000		3000		3000	
Pressure at engine during start, min. (alarm) (20°C)	kPa	1500		1500		1500		1500		1500		1500	
Low pressure limit in starting air receiver	kPa	1600		1600		1600		1600		1600		1600	
Starting air consumption, start (successful)	Nm^3	4.7		4.7		4.7		4.7		4.7		4.7	
Consumption per start (with slowturn)	Nm^3	6.1		6.1		6.1		6.1		6.1		6.1	

Notes:

Note 1 At ISO 15550 conditions (ambient air temperature 25°C, LT-water 25°C) and 100% load. Flow tolerance 5%.

Note 2 At ISO 15550 conditions (ambient air temperature 25°C, LT-water 25°C). Flow tolerance 5% and temperature tolerance 10°C in gas mode operation. Flow tolerance 8% and temperature tolerance 15°C in diesel mode operation.

Note 3 At 100% output and nominal speed. The figures are valid for ambient conditions according to ISO 15550 except for LT-water temperature, which is corresponding to charge air receiver temperature 45ºC in gas operation. With engine driven water and lubricating oil pumps. Tolerance for cooling water heat 10%, tolerance for radiation heat 30%. Fouling factors and a margin to be taken into account when dimensioning heat exchangers.

Note 4 At ambient conditions according to ISO 15550 and receiver temperature 45 °C. Lower calorific value 42 700 kJ/kg for pilot fuel and 49 620 kJ/kg for gas fuel. With engine driven pumps (two cooling water pumps, one lubricating oil pump and pilot fuel pump). Tolerance 5%.

Note 5 Fuel gas pressure given at LHV ≥ 36MJ/m³N. Required fuel gas pressure depends on fuel gas LHV and need to be increased for lower LHV's. Pressure drop in external fuel gas system to be considered. See chapter Fuel system for further information.

Note 6 Minimum pressure for slow turning is 1800kPa.

ME = Engine driving propeller, variable speed

AE = Auxiliary engine driving generator

DE = Diesel-Electric engine driving generator

Subject to revision without notice.

3.2 Wärtsilä 8L34DF

Wärtsilä 8L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Engine speed	rpm	720		750		720		750		750		750	
Engine output	kW	3840		4000		3840		4000		4000		4000	
Mean effective pressure	MPa	2.2		2.2		2.2		2.2		2.2		2.2	
Speed mode		Constant		Constant		Constant		Constant		Constant		Variable	
IMO compliance		Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2
Combustion air system (Note 1)													
Flow at 100% load	kg/s	5.9	7.1	5.9	7.1	5.9	7.1	5.9	7.1	5.9	7.1	5.9	7.3
Temperature at turbocharger intake, max.	°C	45		45		45		45		45		45	
Temperature after air cooler (TE 601), load > 70%	°C	45	-	45	-	45	-	45	-	45	-	45	-
Temperature after air cooler (TE 601), load 30...70%	°C	55	-	55	-	55	-	55	-	55	-	55	-
Temperature after air cooler (TE 601)	°C	-	50	-	50	-	50	-	50	-	50	-	50
Exhaust gas system (Note 2)													
Flow at 100% load	kg/s	6.1	7.4	6.1	7.4	6.1	7.4	6.1	7.4	6.1	7.4	6.1	7.5
Flow at 75% load	kg/s	5.1	5.9	5.1	5.9	5.1	5.9	5.1	5.9	5.1	5.9	5.0	5.8
Flow at 50% load	kg/s	4.1	4.1	4.1	4.1	4.1	4.1	4.1	4.1	4.1	4.1	4.0	4.2
Temperature after turbocharger at 100% load (TE 517)	°C	362	355	381	381	362	346	381	370	381	370	381	362
Temperature after turbocharger at 75% load (TE 517)	°C	383	327	401	349	383	318	401	340	401	340	386	349
Temperature after turbocharger at 50% load (TE 517)	°C	386	350	402	371	386	346	402	366	402	360	341	333
Backpressure, max.	kPa	4		4		4		4		4		4	
Calculated exhaust diameter for 35 m/s	mm	629	688	638	702	629	683	638	696	638	696	638	700
Heat balance at 100% load (Note 3)													
Jacket water, HT-circuit	kW	476	547	496	573	476	542	496	567	496	567	497	591
Charge air, HT-circuit	kW	801	1244	801	1244	801	1244	801	1244	801	1244	801	1288
Charge air, LT-circuit	kW	228	238	228	238	228	238	228	238	228	238	228	245
Lubricating oil, LT-circuit	kW	333	336	346	352	333	333	347	348	347	348	347	375
Radiation	kW	154	156	160	163	154	155	160	162	160	162	160	164
Fuel consumption (Note 4)													
Total energy consumption at 100% load	kJ/kWh	7470	-	7470	-	7470	-	7470	-	7470	-	7470	-
Total energy consumption at 85% load	kJ/kWh	7620	-	7620	-	7620	-	7620	-	7620	-	7570	-
Total energy consumption at 75% load	kJ/kWh	7850	-	7850	-	7850	-	7850	-	7850	-	7590	-
Total energy consumption at 50% load	kJ/kWh	8600	-	8600	-	8600	-	8600	-	8600	-	7790	-
Fuel gas consumption at 100% load	kJ/kWh	7387	-	7387	-	7387	-	7387	-	7387	-	7387	-

Wärtsilä 8L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Fuel gas consumption at 85% load	kJ/kWh	7527	-	7527	-	7527	-	7527	-	7527	-	7471	-
Fuel gas consumption at 75% load	kJ/kWh	7743	-	7743	-	7743	-	7743	-	7743	-	7478	-
Fuel gas consumption at 50% load	kJ/kWh	8435	-	8435	-	8435	-	8435	-	8435	-	7643	-
Fuel oil consumption at 100% load	g/kWh	1.9	191	1.9	192	1.9	189	1.9	190	1.9	190	1.9	190
Fuel oil consumption at 85% load	g/kWh	2.2	188	2.2	189	2.2	186	2.2	187	2.2	187	2.2	186
Fuel oil consumption at 75% load	g/kWh	2.5	188	2.5	189	2.5	186	2.5	187	2.5	187	2.5	184
Fuel oil consumption 50% load	g/kWh	3.8	194	3.8	195	3.8	194	3.8	195	3.8	195	3.4	183
Fuel gas system (Note 5)													
Gas pressure at engine inlet, min (PT901)	kPa (a)	535	-	535	-	535	-	535	-	535	-	535	-
Gas pressure to Gas Valve Unit, min	kPa (a)	655	-	655	-	655	-	655	-	655	-	655	-
Gas temperature before Gas Valve Unit	°C	0...60	-	0...60	-	0...60	-	0...60	-	0...60	-	0...60	-
Fuel oil system													
Pressure before injection pumps (PT 101)	kPa	700±50		700±50		700±50		700±50		700±50		700±50	
Fuel oil flow to engine, approx	m^3/h	4.1		4.3		4.1		4.3		4.3		4.3	
HFO viscosity before the engine	cSt	-	16...24	-	16...24	-	16...24	-	16...24	-	16...24	-	16...24
Max. HFO temperature before engine (TE 101)	°C	-	140	-	140	-	140	-	140	-	140	-	140
MDF viscosity, min.	cSt	2.0		2.0		2.0		2.0		2.0		2.0	
Max. MDF temperature before engine (TE 101)	°C	45		45		45		45		45		45	
Leak fuel quantity (HFO), clean fuel at 100% load	kg/h		3.0		3.1		3.0		3.1		3.1		3.1
Leak fuel quantity (MDF), clean fuel at 100% load	kg/h	7.4	14.8	7.8	15.5	7.4	14.8	7.8	15.5	7.8	15.5	7.8	15.7
Pilot fuel (MDF) viscosity before the engine	cSt	2...11		2...11		2...11		2...11		2...11		2...11	
Pilot fuel pressure at engine inlet (PT 112)	kPa (a)	550...750		550...750		550...750		550...750		550...750		550...750	
Pilot fuel pressure drop after engine, max	kPa	150		150		150		150		150		150	
Pilot fuel return flow at 100% load	kg/h	635		635		590		590		590		590	
Lubricating oil system													
Pressure before bearings, nom. (PT 201)	kPa	500		500		500		500		500		500	
Suction ability, including pipe loss, max.	kPa	30		30		30		30		30		30	
Priming pressure, nom. (PT 201)	kPa	50		50		50		50		50		50	
Suction ability priming pump, including pipe loss, max.	kPa	30		30		30		30		30		30	

Wärtsilä 8L34DF		AUX	AUX	DE	DE	ME	ME
		Gas mode / Diesel mode	Gas mode / Diesel mode	Gas mode / Diesel mode	Gas mode / Diesel mode	Gas mode / Diesel mode	Gas mode / Diesel mode
Cylinder output	**kW**	**480**	**500**	**480**	**500**	**500**	**500**
Temperature before bearings, nom. (TE 201)	°C	63	63	63	63	63	63
Temperature after engine, approx.	°C	78	78	78	78	78	78
Pump capacity (main), engine driven	m³/h	101	105	101	105	105	105
Pump capacity (main), electrically driven	m³/h	91	95	91	95	95	95
Priming pump capacity (50/60Hz)	m³/h	21.6 / 25.9	21.6 / 25.9	21.6 / 25.9	21.6 / 25.9	21.6 / 25.9	21.6 / 25.9
Oil volume, wet sump, nom.	m³	2.0	2.0	2.0	2.0	2.0	2.0
Oil volume in separate system oil tank	m³	4	4	4	4	4	4
Oil consumption at 100% load, approx.	g/kWh	0.4	0.4	0.4	0.4	0.4	0.4
Crankcase ventilation flow rate at full load	l/min	1120	1120	1120	1120	1120	1120
Crankcase ventilation backpressure, max.	kPa	0.3	0.3	0.3	0.3	0.3	0.3
Oil volume in turning device	l	8.5...9.5	8.5...9.5	8.5...9.5	8.5...9.5	8.5...9.5	8.5...9.5
Oil volume in speed governor	l	1.4...2.2	1.4...2.2	1.4...2.2	1.4...2.2	1.4...2.2	1.4...2.2
HT cooling water system							
Pressure at engine, after pump, nom. (PT 401)	kPa	250 + static	250 + static	250 + static	250 + static	250 + static	250 + static
Pressure at engine, after pump, max. (PT 401)	kPa	530	530	530	530	530	530
Temperature before cylinders, approx. (TE 401)	°C	85	85	85	85	85	85
Temperature after engine, nom.	°C	96	96	96	96	96	96
Capacity of engine driven pump, nom.	m³/h	75	75	75	75	75	80
Pressure drop over engine, total	kPa	100	100	100	100	100	100
Pressure drop in external system, max.	kPa	100	100	100	100	100	100
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150	70...150
Water volume in engine	m³	0.51	0.51	0.51	0.51	0.51	0.51
Delivery head of stand-by pump	kPa	250	250	250	250	250	250
LT cooling water system							
Pressure at engine, after pump, nom. (PT 471)	kPa	250+ static	250+ static	250+ static	250+ static	250+ static	250+ static
Pressure at engine, after pump, max. (PT 471)	kPa	530	530	530	530	530	530
Temperature before engine, max. (TE 471)	°C	38	38	38	38	38	38
Temperature before engine, min. (TE 471)	°C	25	25	25	25	25	25
Capacity of engine driven pump, nom.	m³/h	75	75	75	75	75	80
Pressure drop over charge air cooler	kPa	35	35	35	35	35	35

Wärtsilä 8L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Pressure drop in external system, max.	kPa	100		100		100		100		100		100	
Pressure from expansion tank	kPa	70...150		70...150		70...150		70...150		70...150		70...150	
Delivery head of stand-by pump	kPa	250		250		250		250		250		250	
Starting air system (Note 6)													
Pressure, nom.	kPa	3000		3000		3000		3000		3000		3000	
Pressure, max.	kPa	3000		3000		3000		3000		3000		3000	
Pressure at engine during start, min. (alarm) (20°C)	kPa	1500		1500		1500		1500		1500		1500	
Low pressure limit in starting air receiver	kPa	1600		1600		1600		1600		1600		1600	
Starting air consumption, start (successful)	Nm^3	5.7		5.7		5.7		5.7		5.7		5.7	
Consumption per start (with slowturn)	Nm^3	7.4		7.4		7.4		7.4		7.4		7.4	

Notes:

Note 1 At ISO 15550 conditions (ambient air temperature 25°C, LT-water 25°C) and 100% load. Flow tolerance 5%.

Note 2 At ISO 15550 conditions (ambient air temperature 25°C, LT-water 25°C). Flow tolerance 5% and temperature tolerance 10°C in gas mode operation. Flow tolerance 8% and temperature tolerance 15°C in diesel mode operation.

Note 3 At 100% output and nominal speed. The figures are valid for ambient conditions according to ISO 15550 except for LT-water temperature, which is corresponding to charge air receiver temperature 45ºC in gas operation. With engine driven water and lubricating oil pumps. Tolerance for cooling water heat 10%, tolerance for radiation heat 30%. Fouling factors and a margin to be taken into account when dimensioning heat exchangers.

Note 4 At ambient conditions according to ISO 15550 and receiver temperature 45 °C. Lower calorific value 42 700 kJ/kg for pilot fuel and 49 620 kJ/kg for gas fuel. With engine driven pumps (two cooling water pumps, one lubricating oil pump and pilot fuel pump). Tolerance 5%.

Note 5 Fuel gas pressure given at LHV ≥ 36MJ/m³N. Required fuel gas pressure depends on fuel gas LHV and need to be increased for lower LHV's. Pressure drop in external fuel gas system to be considered. See chapter Fuel system for further information.

Note 6 Minimum pressure for slow turning is 1800kPa.

ME = Engine driving propeller, variable speed

AE = Auxiliary engine driving generator

DE = Diesel-Electric engine driving generator

Subject to revision without notice.

3.3 Wärtsilä 9L34DF

Wärtsilä 9L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Engine speed	rpm	720		750		720		750		750		750	
Engine output	kW	4320		4500		4320		4500		4500		4500	
Mean effective pressure	MPa	2.2		2.2		2.2		2.2		2.2		2.2	
Speed mode		Constant		Constant		Constant		Constant		Constant		Variable	
IMO compliance		Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2
Combustion air system (Note 1)													
Flow at 100% load	kg/s	6.7	8.0	6.7	8.0	6.7	8.0	6.7	8.0	6.7	8.0	6.7	8.2
Temperature at turbocharger intake, max.	°C	45		45		45		45		45		45	
Temperature after air cooler (TE 601), load > 70%	°C	45	-	45	-	45	-	45	-	45	-	45	-
Temperature after air cooler (TE 601), load 30...70%	°C	55	-	55	-	55	-	55	-	55	-	55	-
Temperature after air cooler (TE 601)	°C	-	50	-	50	-	50	-	50	-	50	-	50
Exhaust gas system (Note 2)													
Flow at 100% load	kg/s	6.8	8.3	6.8	8.3	6.8	8.3	6.8	8.3	6.8	8.3	6.8	8.5
Flow at 75% load	kg/s	5.8	6.7	5.8	6.7	5.8	6.7	5.8	6.7	5.8	6.7	5.6	6.5
Flow at 50% load	kg/s	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.5	4.7
Temperature after turbocharger at 100% load (TE 517)	°C	362	355	381	381	362	346	381	370	381	370	381	361
Temperature after turbocharger at 75% load (TE 517)	°C	383	327	401	349	383	318	401	340	401	340	386	348
Temperature after turbocharger at 50% load (TE 517)	°C	386	350	402	371	386	346	402	366	402	366	341	333
Backpressure, max.	kPa	4		4		4		4		4		4	
Calculated exhaust diameter for 35 m/s	mm	667	730	677	745	667	724	677	739	677	739	677	741
Heat balance at 100% load (Note 3)													
Jacket water, HT-circuit	kW	535	616	557	645	535	609	557	638	557	638	559	664
Charge air, HT-circuit	kW	901	1399	901	1399	901	1399	901	1399	901	1399	901	1449
Charge air, LT-circuit	kW	257	268	257	268	257	268	257	268	257	268	257	275
Lubricating oil, LT-circuit	kW	374	378	389	396	374	374	390	392	390	392	390	422
Radiation	kW	173	176	180	184	173	174	180	182	180	182	180	184
Fuel consumption (Note 4)													
Total energy consumption at 100% load	kJ/kWh	7470	-	7470	-	7470	-	7470	-	7470	-	7470	-
Total energy consumption at 85% load	kJ/kWh	7620	-	7620	-	7620	-	7620	-	7620	-	7570	-
Total energy consumption at 75% load	kJ/kWh	7850	-	7850	-	7850	-	7850	-	7850	-	7590	-
Total energy consumption at 50% load	kJ/kWh	8600	-	8600	-	8600	-	8600	-	8600	-	7790	-
Fuel gas consumption at 100% load	kJ/kWh	7387	-	7387	-	7387	-	7387	-	7387	-	7387	-

3.4 Wärtsilä 12V34DF

Wärtsilä 12V34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Engine speed	rpm	720		750		720		750		750		750	
Engine output	kW	5760		6000		5760		6000		6000		6000	
Mean effective pressure	MPa	2.2		2.2		2.2		2.2		2.2		2.2	
Speed mode		Constant		Constant		Constant		Constant		Constant		Variable	
IMO compliance		Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2
Combustion air system (Note 1)													
Flow at 100% load	kg/s	8.9	10.7	8.9	10.7	8.9	10.7	8.9	10.7	8.9	10.7	8.9	11.0
Temperature at turbocharger intake, max.	°C	45		45		45		45		45		45	
Temperature after air cooler (TE 601), load > 70%	°C	45	-	45	-	45	-	45	-	45	-	45	-
Temperature after air cooler (TE 601), load 30...70%	°C	55	-	55	-	55	-	55	-	55	-	55	-
Temperature after air cooler (TE 601)	°C	-	50	-	50	-	50	-	50	-	50	-	50
Exhaust gas system (Note 2)													
Flow at 100% load	kg/s	9.1	11.0	9.1	11.0	9.1	11.0	9.1	11.0	9.1	11.0	9.1	11.3
Flow at 75% load	kg/s	7.7	8.9	7.7	8.9	7.7	8.9	7.7	8.9	7.7	8.9	7.4	8.6
Flow at 50% load	kg/s	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.0	6.2
Temperature after turbocharger at 100% load (TE 517)	°C	357	351	376	375	357	341	376	365	376	365	377	356
Temperature after turbocharger at 75% load (TE 517)	°C	378	323	397	344	378	314	397	335	397	335	382	344
Temperature after turbocharger at 50% load (TE 517)	°C	382	346	398	366	382	342	398	362	398	362	337	329
Backpressure, max.	kPa	4		4		4		4		4		4	
Calculated exhaust diameter for 35 m/s	mm	767	840	778	856	767	833	778	850	778	850	779	853
Heat balance at 100% load (Note 3)													
Jacket water, HT-circuit	kW	710	817	739	855	710	808	739	846	739	846	741	881
Charge air, HT-circuit	kW	1201	1866	1201	1866	1201	1866	1201	1866	1201	1866	1195	1932
Charge air, LT-circuit	kW	343	357	343	357	343	357	343	357	343	357	343	367
Lubricating oil, LT-circuit	kW	496	502	517	525	496	497	517	520	517	520	518	560
Radiation	kW	230	233	239	244	230	231	239	241	239	241	239	244
Fuel consumption (Note 4)													
Total energy consumption at 100% load	kJ/kWh	7430	-	7430	-	7430	-	7430	-	7430	-	7430	-
Total energy consumption at 85% load	kJ/kWh	7590	-	7590	-	7590	-	7590	-	7590	-	7530	-
Total energy consumption at 75% load	kJ/kWh	7810	-	7810	-	7810	-	7810	-	7810	-	7550	-
Total energy consumption at 50% load	kJ/kWh	8550	-	8550	-	8550	-	8550	-	8550	-	7750	-
Fuel gas consumption at 100% load	kJ/kWh	7349	-	7349	-	7349	-	7349	-	7349	-	7349	-

Wärtsilä 12V34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Fuel gas consumption at 85% load	kJ/kWh	7488	-	7488	-	7488	-	7488	-	7488	-	7431	-
Fuel gas consumption at 75% load	kJ/kWh	7702	-	7702	-	7702	-	7702	-	7702	-	7438	-
Fuel gas consumption at 50% load	kJ/kWh	8390	-	8390	-	8390	-	8390	-	8390	-	7607	-
Fuel oil consumption at 100% load	g/kWh	1.9	190	1.9	191	1.9	188	1.9	189	1.9	189	1.9	189
Fuel oil consumption at 85% load	g/kWh	2.2	187	2.2	188	2.2	185	2.2	186	2.2	186	2.2	185
Fuel oil consumption at 75% load	g/kWh	2.5	187	2.5	188	2.5	185	2.5	186	2.5	186	2.5	183
Fuel oil consumption 50% load	g/kWh	3.8	193	3.8	194	3.8	193	3.8	194	3.8	194	3.4	182
Fuel gas system (Note 5)													
Gas pressure at engine inlet, min (PT901)	kPa (a)	535	-	535	-	535	-	535	-	535	-	535	-
Gas pressure to Gas Valve Unit, min	kPa (a)	655	-	655	-	655	-	655	-	655	-	655	-
Gas temperature before Gas Valve Unit	°C	0...60	-	0...60	-	0...60	-	0...60	-	0...60	-	0...60	-
Fuel oil system													
Pressure before injection pumps (PT 101)	kPa	700±50		700±50		700±50		700±50		700±50		700±50	
Fuel oil flow to engine, approx	m³/h	6.2		6.5		6.1		6.4		6.4		6.4	
HFO viscosity before the engine	cSt	-	16...24	-	16...24	-	16...24	-	16...24	-	16...24	-	16...24
Max. HFO temperature before engine (TE 101)	°C	-	140	-	140	-	140	-	140	-	140	-	140
MDF viscosity, min.	cSt	2.0		2.0		2.0		2.0		2.0		2.0	
Max. MDF temperature before engine (TE 101)	°C	45		45		45		45		45		45	
Leak fuel quantity (HFO), clean fuel at 100% load	kg/h		4.4		4.6		4.4		4.6		4.6		4.7
Leak fuel quantity (MDF), clean fuel at 100% load	kg/h	11.1	22.1	11.6	23.2	11.1	22.1	11.6	23.2	11.6	23.2	11.7	23.4
Pilot fuel (MDF) viscosity before the engine	cSt	2...11		2...11		2...11		2...11		2...11		2...11	
Pilot fuel pressure at engine inlet (PT 112)	kPa (a)	550...750		550...750		550...750		550...750		550...750		550...750	
Pilot fuel pressure drop after engine, max	kPa	150		150		150		150		150		150	
Pilot fuel return flow at 100% load	kg/h	680		680		680		680		680		680	
Lubricating oil system													
Pressure before bearings, nom. (PT 201)	kPa	500		500		500		500		500		500	
Suction ability, including pipe loss, max.	kPa	40		40		40		40		40		40	
Priming pressure, nom. (PT 201)	kPa	50		50		50		50		50		50	
Suction ability priming pump, including pipe loss, max.	kPa	35		35		35		35		35		35	
Temperature before bearings, nom. (TE 201)	°C	63		63		63		63		63		63	
Temperature after engine, approx.	°C	81		81		81		81		81		81	
Pump capacity (main), engine driven	m³/h	124		129		124		129		129		129	

Wärtsilä 12V34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Pump capacity (main), electrically driven	m^3/h	106		110		106		110		110		110	
Priming pump capacity (50/60Hz)	m^3/h	38.0 / 45.9		38.0 / 45.9		38.0 / 45.9		38.0 / 45.9		38.0 / 45.9		38.0 / 45.9	
Oil volume, wet sump, nom.	m^3	3.0		3.0		3.0		3.0		3.0		3.0	
Oil volume in separate system oil tank	m^3	6		6		6		6		6		6	
Oil consumption at 100% load, approx.	g/kWh	0.4		0.4		0.4		0.4		0.4		0.4	
Crankcase ventilation flow rate at full load	l/min	1680		1680		1680		1680		1680		1680	
Crankcase ventilation backpressure, max.	kPa	0.3		0.3		0.3		0.3		0.3		0.3	
Oil volume in turning device	l	...		...		...		...		...		...	
Oil volume in speed governor	l	1.4...2.2		1.4...2.2		1.4...2.2		1.4...2.2		1.4...2.2		1.4...2.2	
HT cooling water system													
Pressure at engine, after pump, nom. (PT 401)	kPa	250 + static		250 + static		250 + static		250 + static		250 + static		250 + static	
Pressure at engine, after pump, max. (PT 401)	kPa	530		530		530		530		530		530	
Temperature before cylinders, approx. (TE 401)	°C	85		85		85		85		85		85	
Temperature after engine, nom.	°C	96		96		96		96		96		96	
Capacity of engine driven pump, nom.	m^3/h	100		100		100		100		100		100	
Pressure drop over engine, total	kPa	100		100		100		100		100		100	
Pressure drop in external system, max.	kPa	100		100		100		100		100		100	
Pressure from expansion tank	kPa	70...150		70...150		70...150		70...150		70...150		70...150	
Water volume in engine	m^3	0.74		0.74		0.74		0.74		0.74		0.74	
Delivery head of stand-by pump	kPa	250		250		250		250		250		250	
LT cooling water system													
Pressure at engine, after pump, nom. (PT 471)	kPa	250+ static		250+ static		250+ static		250+ static		250+ static		250+ static	
Pressure at engine, after pump, max. (PT 471)	kPa	530		530		530		530		530		530	
Temperature before engine, max. (TE 471)	°C	38		38		38		38		38		38	
Temperature before engine, min. (TE 471)	°C	25		25		25		25		25		25	
Capacity of engine driven pump, nom.	m^3/h	100		100		100		100		100		100	
Pressure drop over charge air cooler	kPa	35		35		35		35		35		35	
Pressure drop in external system, max.	kPa	100		100		100		100		100		100	
Pressure from expansion tank	kPa	70...150		70...150		70...150		70...150		70...150		70...150	
Delivery head of stand-by pump	kPa	250		250		250		250		250		250	
Starting air system (Note 6)													
Pressure, nom.	kPa	3000		3000		3000		3000		3000		3000	
Pressure, max.	kPa	3000		3000		3000		3000		3000		3000	

Wärtsilä 12V34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Pressure at engine during start, min. (alarm) (20°C)	kPa	1500		1500		1500		1500		1500		1500	
Low pressure limit in starting air receiver	kPa	1600		1600		1600		1600		1600		1600	
Starting air consumption, start (successful)	Nm^3	6.8		6.8		6.8		6.8		6.8		6.8	
Consumption per start at (with slowturn)	Nm^3	8.8		8.8		8.8		8.8		8.8		8.8	

Notes:

Note 1 At ISO 15550 conditions (ambient air temperature 25°C, LT-water 25°C) and 100% load. Flow tolerance 5%.

Note 2 At ISO 15550 conditions (ambient air temperature 25°C, LT-water 25°C). Flow tolerance 5% and temperature tolerance 10°C in gas mode operation. Flow tolerance 8% and temperature tolerance 15°C in diesel mode operation.

Note 3 At 100% output and nominal speed. The figures are valid for ambient conditions according to ISO 15550 except for LT-water temperature, which is corresponding to charge air receiver temperature 45ºC in gas operation. With engine driven water and lubricating oil pumps. Tolerance for cooling water heat 10%, tolerance for radiation heat 30%. Fouling factors and a margin to be taken into account when dimensioning heat exchangers.

Note 4 At ambient conditions according to ISO 15550 and receiver temperature 45 °C. Lower calorific value 42 700 kJ/kg for pilot fuel and 49 620 kJ/kg for gas fuel. With engine driven pumps (two cooling water pumps, one lubricating oil pump and pilot fuel pump). Tolerance 5%.

Note 5 Fuel gas pressure given at LHV ≥ 36MJ/m³N. Required fuel gas pressure depends on fuel gas LHV and need to be increased for lower LHV's. Pressure drop in external fuel gas system to be considered. See chapter Fuel system for further information.

Note 6 Minimum pressure for slow turning is 1800kPa.

ME = Engine driving propeller, variable speed

AE = Auxiliary engine driving generator

DE = Diesel-Electric engine driving generator

Subject to revision without notice.

1.6.2 Generating sets

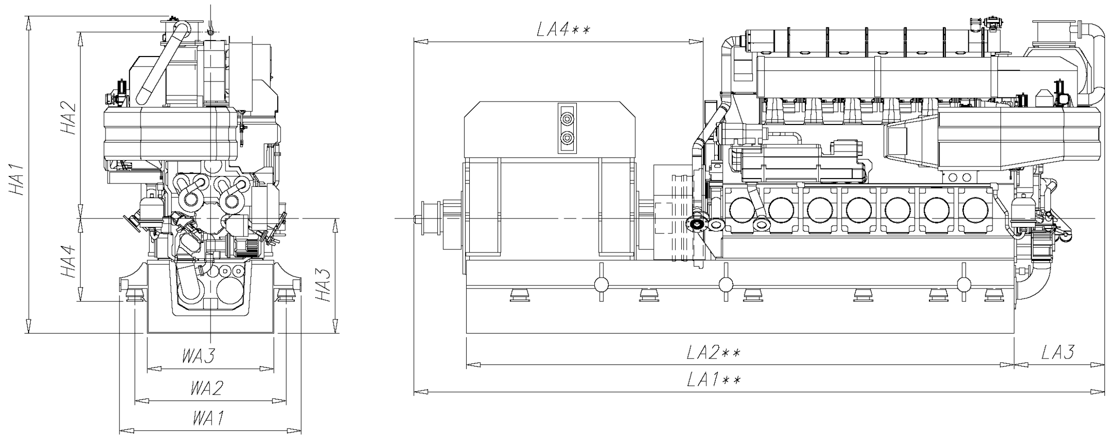


Fig 1-5 **In-line engines (DAAE082427)**

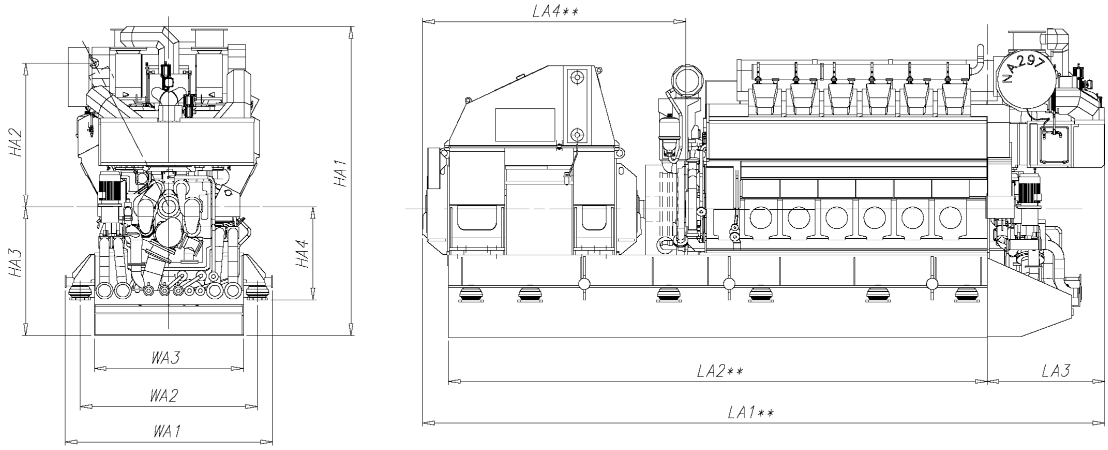


Fig 1-6 **V engines (DAAE082975)**

Engine	kW/cyl	LA1**	LA2**	LA3	LA4**	WA1	WA2	WA3	HA1	HA2	HA3	HA4	Weight**
W 6L34DF	480	8765	6900	1215	3160	2290	1910	1600	4000	2345	1450	1055	60
Wärtsilä 8L34DF	480	10410	8650	1285	3645	2690	2310	2000	4180	2345	1630	1055	76
W 9L34DF	480	10610	8850	1285	3845	2890	2510	2200	4180	2345	1630	1055	87
W 12V34DF	480	10260	7950	1985	3775	3060	2620	2200	4335	2120	1900	1375	99
W 16V34DF	480	11465	9130	1925	3765	3360	2920	2500	4445	2120	1850	1375	124

** Dependent on generator and flexible coupling.

All dimensions in mm. Weight in metric tons with liquids.

3.3 Wärtsilä 9L34DF

Wärtsilä 9L34DF		AUX		AUX		DE		DE		ME		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	**kW**	**480**		**500**		**480**		**500**		**500**		**500**	
Engine speed	rpm	720		750		720		750		750		750	
Engine output	kW	4320		4500		4320		4500		4500		4500	
Mean effective pressure	MPa	2.2		2.2		2.2		2.2		2.2		2.2	
Speed mode		Constant		Constant		Constant		Constant		Constant		Variable	
IMO compliance		Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2	Tier 3	Tier 2
Combustion air system (Note 1)													
Flow at 100% load	kg/s	6.7	8.0	6.7	8.0	6.7	8.0	6.7	8.0	6.7	8.0	6.7	8.2
Temperature at turbocharger intake, max.	°C	45		45		45		45		45		45	
Temperature after air cooler (TE 601), load > 70%	°C	45	-	45	-	45	-	45	-	45	-	45	-
Temperature after air cooler (TE 601), load 30...70%	°C	55	-	55	-	55	-	55	-	55	-	55	-
Temperature after air cooler (TE 601)	°C	-	50	-	50	-	50	-	50	-	50	-	50
Exhaust gas system (Note 2)													
Flow at 100% load	kg/s	6.8	8.3	6.8	8.3	6.8	8.3	6.8	8.3	6.8	8.3	6.8	8.5
Flow at 75% load	kg/s	5.8	6.7	5.8	6.7	5.8	6.7	5.8	6.7	5.8	6.7	5.6	6.5
Flow at 50% load	kg/s	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.5	4.7
Temperature after turbocharger at 100% load (TE 517)	°C	362	355	381	381	362	346	381	370	381	370	381	361
Temperature after turbocharger at 75% load (TE 517)	°C	383	327	401	349	383	318	401	340	401	340	386	348
Temperature after turbocharger at 50% load (TE 517)	°C	386	350	402	371	386	346	402	366	402	366	341	333
Backpressure, max.	kPa	4		4		4		4		4		4	
Calculated exhaust diameter for 35 m/s	mm	667	730	677	745	667	724	677	739	677	739	677	741
Heat balance at 100% load (Note 3)													
Jacket water, HT-circuit	kW	535	616	557	645	535	609	557	638	557	638	559	664
Charge air, HT-circuit	kW	901	1399	901	1399	901	1399	901	1399	901	1399	901	1449
Charge air, LT-circuit	kW	257	268	257	268	257	268	257	268	257	268	257	275
Lubricating oil, LT-circuit	kW	374	378	389	396	374	374	390	392	390	392	390	422
Radiation	kW	173	176	180	184	173	174	180	182	180	182	180	184
Fuel consumption (Note 4)													
Total energy consumption at 100% load	kJ/kWh	7470	-	7470	-	7470	-	7470	-	7470	-	7470	-
Total energy consumption at 85% load	kJ/kWh	7620	-	7620	-	7620	-	7620	-	7620	-	7570	-
Total energy consumption at 75% load	kJ/kWh	7850	-	7850	-	7850	-	7850	-	7850	-	7590	-
Total energy consumption at 50% load	kJ/kWh	8600	-	8600	-	8600	-	8600	-	8600	-	7790	-
Fuel gas consumption at 100% load	kJ/kWh	7387	-	7387	-	7387	-	7387	-	7387	-	7387	-

11.2 Exhaust gas outlet

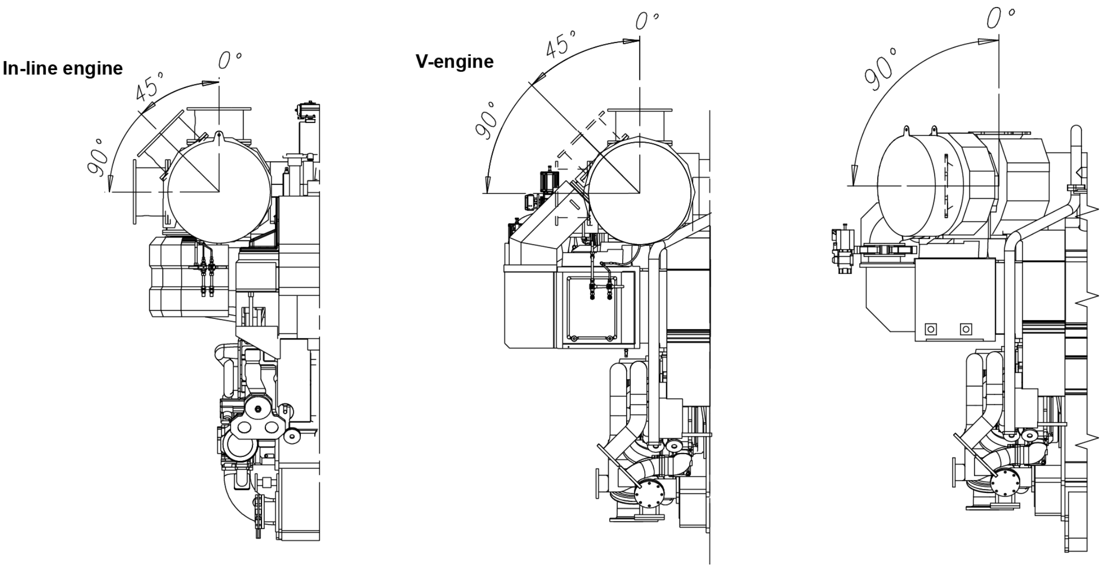
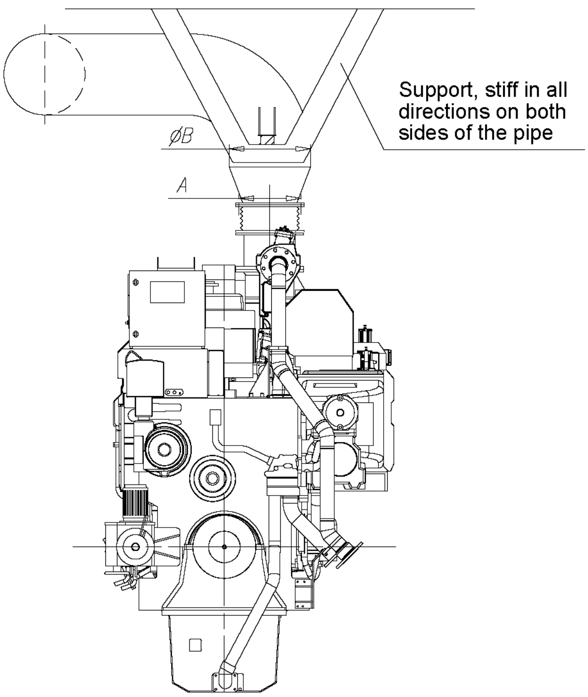


Fig 11-3 Exhaust pipe connections (DAAF068270A)

Engine	TC type	TC in free end	TC in driving end
W 6L34DF	NT1-10	0°	-
	A145	0°, 45°, 90°	0°, 45°, 90°
W 8L34DF	NT1-10	0°	-
W 9L34DF	A155	0°, 45°, 90°	0°, 45°, 90°
W 12V34DF	NT1-10	0°	-
W 16V34DF	NT1-10	0°	-



Engine	TC type	A	ØB [mm]
W 6L34DF	NT1-10A	DN500	900
	A145 A155	DN350	600
W 8L34DF	NT1-10A	DN500	900
W 9L34DF	A145 A155	DN450	700

Fig 11-4 Exhaust pipe, diameters and support (DAAF068269)

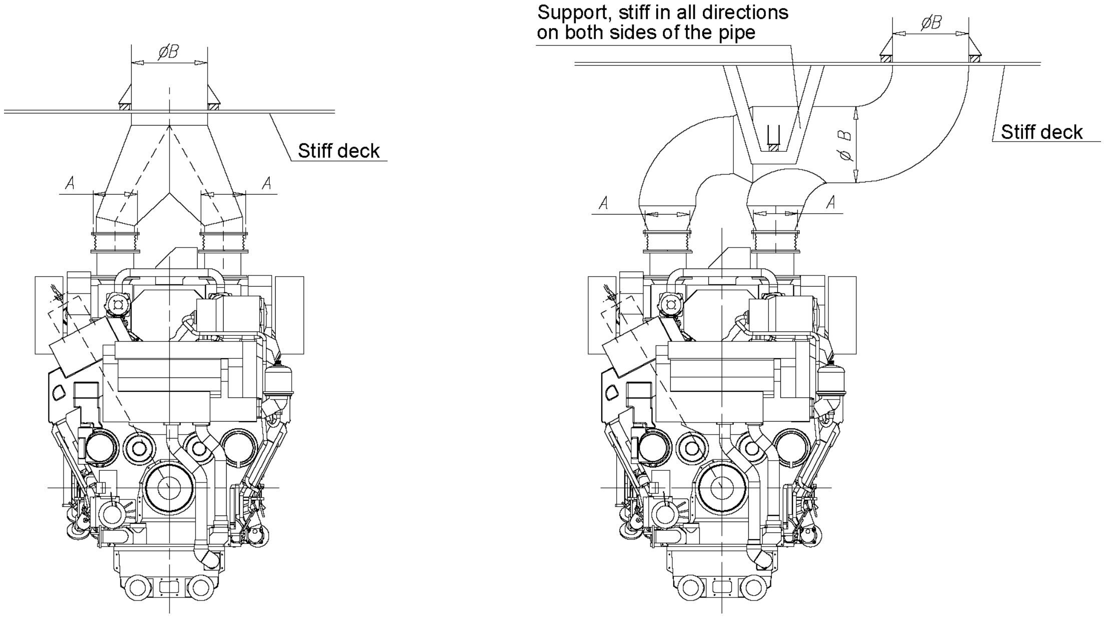


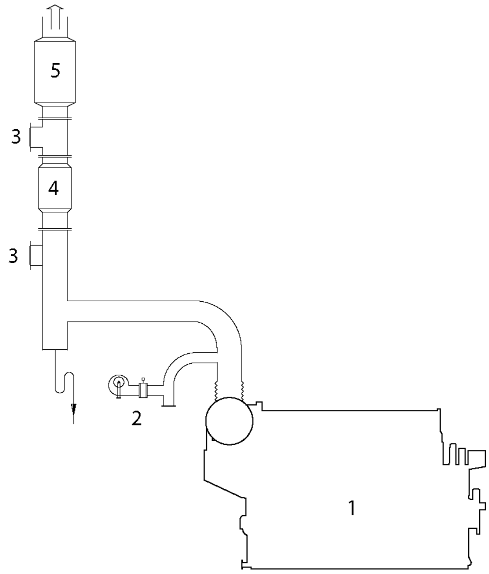
Fig 11-5 Exhaust pipe, diameters and support (DAAF068200A, -04A)

Engine	TC type	A	ØB [mm]
W 12V34DF	NT1-10A	DN500	900
W 16V34DF	NT1-10A	DN500	900

11.3 External exhaust gas system

Each engine should have its own exhaust pipe into open air. Backpressure, thermal expansion and supporting are some of the decisive design factors.

Flexible bellows must be installed directly on the turbocharger outlet, to compensate for thermal expansion and prevent damages to the turbocharger due to vibrations.



1 Duel Fuel engine

2 Exhaust gas ventilation unit

3 Rupture discs

4 Exhaust gas boiler

5 Silencer

Fig 11-6 External exhaust gas system

11.3.1 System design - safety aspects

Natural gas may enter the exhaust system if a malfunction occurs during gas operation. The gas may accumulate in the exhaust piping and it could be ignited in case a source of ignition (such as a spark) appears in the system. The external exhaust system must therefore be designed so that the pressure build-up in case of an explosion does not exceed the maximum permissible pressure for any of the components in the system. The engine can tolerate a pressure of at least 200 kPa. Other components in the system might have a lower maximum pressure limit. The consequences of a possible gas explosion can be minimized with proper design of the exhaust system; the engine will not be damaged and the explosion gases will be safely directed through predefined routes. The following guidelines should be observed, when designing the external exhaust system:

- The piping and all other components in the exhaust system should have a constant upward slope to prevent gas from accumulating in the system. If horizontal pipe sections cannot be completely avoided, their length should be kept to a minimum. The length of a single horizontal pipe section should not exceed five times the diameter of the pipe. Silencers and exhaust boilers etc. must be designed so that gas cannot accumulate inside.
- The exhaust system must be equipped with explosion relief devices, such as rupture discs, in order to ensure safe discharge of explosion pressure. The outlets from explosion relief devices must be in locations where the pressure can be safely released.

In addition the control and automation systems include the following safety functions:

- Before start the engine is automatically ventilated, i.e. rotated without injecting any fuel.
- During the start sequence, before activating the gas admission to the engine, an automatic combustion check is performed to ensure that the pilot fuel injection system is working correctly.

13. Exhaust Emissions

Exhaust emissions from the dual fuel engine mainly consist of nitrogen, carbon dioxide (CO2) and water vapour with smaller quantities of carbon monoxide (CO), sulphur oxides (SOx) and nitrogen oxides (NOx), partially reacted and non-combusted hydrocarbons and particulates.

13.1 Dual fuel engine exhaust components

Due to the high efficiency and the clean fuel used in a dual fuel engine in gas mode, the exhaust gas emissions when running on gas are extremely low. In a dual fuel engine, the air-fuel ratio is very high, and uniform throughout the cylinders. Maximum temperatures and subsequent NOx formation are therefore low, since the same specific heat quantity released to combustion is used to heat up a large mass of air. Benefitting from this unique feature of the lean-burn principle, the NOx emissions from the Wärtsilä DF engine is very low, complying with most existing legislation. In gas mode most stringent emissions of IMO, EPA and SECA are met, while in diesel mode the dual fuel engine is a normal diesel engine.

To reach low emissions in gas operation, it is essential that the amount of injected diesel fuel is very small. The Wärtsilä DF engines therefore use a "micro-pilot" with less than 1% diesel fuel injected at nominal load. Thus the emissions of SOx from the dual fuel engine are negligable. When the engine is in diesel operating mode, the emissions are in the same range as for any ordinary diesel engine, and the engine will be delivered with an EIAPP certificate to show compliance with the MARPOL Annex VI.

13.2 Marine exhaust emissions legislation

13.2.1 International Maritime Organization (IMO)

The increasing concern over the air pollution has resulted in the introduction of exhaust emission controls to the marine industry. To avoid the growth of uncoordinated regulations, the IMO (International Maritime Organization) has developed the Annex VI of MARPOL 73/78, which represents the first set of regulations on the marine exhaust emissions.

13.2.1.1 MARPOL Annex VI - Air Pollution

The MARPOL 73/78 Annex VI entered into force 19 May 2005. The Annex VI sets limits on Nitrogen Oxides, Sulphur Oxides and Volatile Organic Compounds emissions from ship exhausts and prohibits deliberate emissions of ozone depleting substances.

13.2.1.1.1 Nitrogen Oxides, NO_x Emissions

The MARPOL 73/78 Annex VI regulation 13, Nitrogen Oxides, applies to diesel engines over 130 kW installed on ships built (defined as date of keel laying or similar stage of construction) on or after January 1, 2000 and different levels (Tiers) of NOx control apply based on the ship construction date. The NO_x emissions limit is expressed as dependent on engine speed. IMO has developed a detailed NO_x Technical Code which regulates the enforcement of these rules.

13.2.1.1.1.1 EIAPP Certification

An EIAPP (Engine International Air Pollution Prevention) Certificate is issued for each engine showing that the engine complies with the NO_x regulations set by the IMO.

When testing the engine for NO_x emissions, the reference fuel is Marine Diesel Oil (distillate) and the test is performed according to ISO 8178 test cycles. Subsequently, the NO_x value has to be calculated using different weighting factors for different loads that have been corrected to ISO 8178 conditions. The used ISO 8178 test cycles are presented in the following table.

Table 13-1 ISO 8178 test cycles

D2: Constant-speed auxiliary engine application	Speed (%)	100	100	100	100	100
	Power (%)	100	75	50	25	10
	Weighting factor	0.05	0.25	0.3	0.3	0.1

E2: Constant-speed main propulsion application including diesel-electric drive and all controllable-pitch propeller installations	Speed (%)	100	100	100	100
	Power (%)	100	75	50	25
	Weighting factor	0.2	0.5	0.15	0.15

C1: Variable -speed and -load auxiliary engine application	Speed	Rated				Intermediate			Idle
	Torque (%)	100	75	50	10	100	75	50	0
	Weighting factor	0.15	0.15	0.15	0.1	0.1	0.1	0.1	0.15

13.2.1.1.1.1.1 Engine family/group

As engine manufacturers have a variety of engines ranging in size and application, the NO_x Technical Code allows the organising of engines into families or groups. By definition, an engine family is a manufacturer's grouping, which through their design, are expected to have similar exhaust emissions characteristics i.e., their basic design parameters are common. When testing an engine family, the engine which is expected to develop the worst emissions is selected for testing. The engine family is represented by the parent engine, and the certification emission testing is only necessary for the parent engine. Further engines can be certified by checking document, component, setting etc., which have to show correspondence with those of the parent engine.

13.2.1.1.1.1.2 Technical file

According to the IMO regulations, a Technical File shall be made for each engine. The Technical File contains information about the components affecting NO_x emissions, and each critical component is marked with a special IMO number. The allowable setting values and parameters for running the engine are also specified in the Technical File. The EIAPP certificate is part of the IAPP (International Air Pollution Prevention) Certificate for the whole ship.

13.2.1.1.1.2 IMO NOx emission standards

The first IMO Tier 1 NOx emission standard entered into force in 2005 and applies to marine diesel engines installed in ships constructed on or after 1.1.2000 and prior to 1.1.2011.

The Marpol Annex VI and the NO_x Technical Code were later undertaken a review with the intention to further reduce emissions from ships and a final adoption for IMO Tier 2 and Tier 3 standards were taken in October 2008.

The IMO Tier 2 NOx standard entered into force 1.1.2011 and replaced the IMO Tier 1 NOx emission standard globally. The Tier 2 NOx standard applies for marine diesel engines installed in ships constructed on or after 1.1.2011.

The IMO Tier 3 NO_x emission standard effective date starts from year 2016. The Tier 3 standard will apply in designated emission control areas (ECA). The ECAs are to be defined by the IMO. So far, the North American ECA and the US Caribbean Sea ECA have been defined and will be effective for marine diesel engines installed in ships constructed on or after 1.1.2016. For other ECAs which might be designated in the future for Tier 3 NOx control, the entry into force date would apply to ships constructed on or after the date of adoption by the MEPC of such an ECA, or a later date as may be specified separately. The IMO Tier 2 NO_x emission standard will apply outside the Tier 3 designated areas.

The NO_x emissions limits in the IMO standards are expressed as dependent on engine speed. These are shown in the following figure.

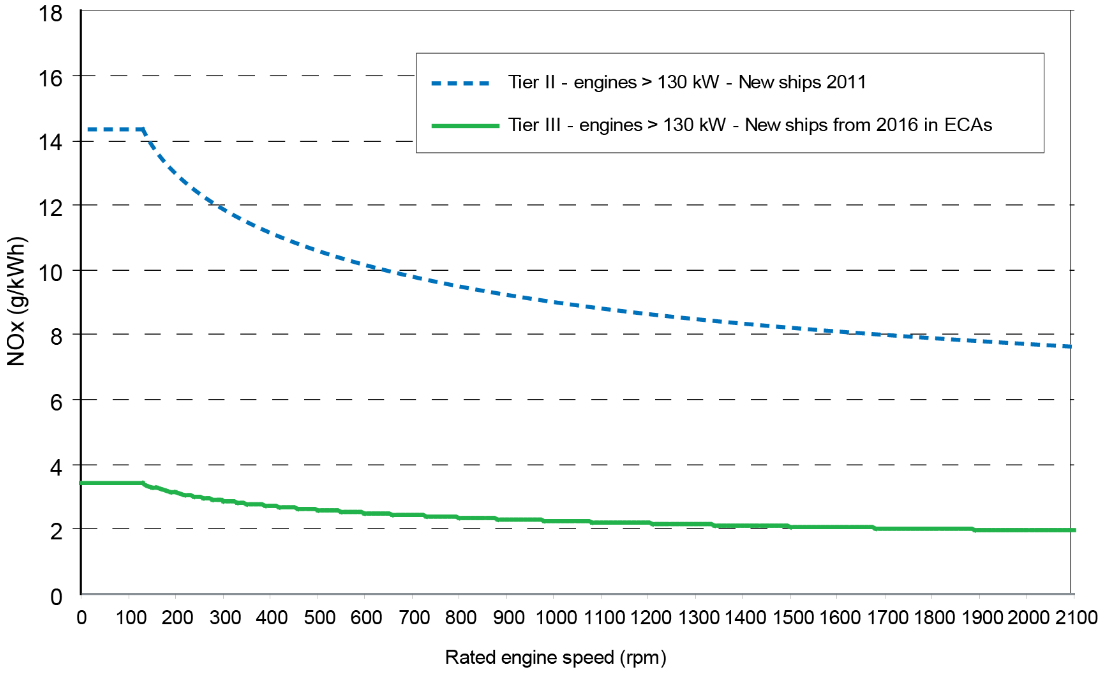


Fig 13-1 IMO NO_x emission limits

13.2.1.1.1.3 IMO Tier 2 NO_x emission standard (new ships 2011)

The IMO Tier 2 NO_x emission standard entered into force in 1.1.2011 and applies globally for new marine diesel engines > 130 kW installed in ships which keel laying date is 1.1.2011 or later.

The IMO Tier 2 NO_x limit is defined as follows:

NO_x [g/kWh] $= 44 \times rpm^{-0.23}$ when $130 < rpm < 2000$

The NO_x level is a weighted average of NO_x emissions at different loads, and the test cycle is based on the engine operating profile according to ISO 8178 test cycles. The IMO Tier 2 NOx level is met by engine internal methods.

13.2.1.1.1.4 IMO Tier 3 NO_x emission standard (new ships from 2016 in ECAs)

The IMO Tier 3 NO_x emission standard will enter into force from year 2016. It will by then apply for new marine diesel engines > 130 kW installed in ships which keel laying date is 1.1.2016 or later when operating inside the North American ECA and the US Caribbean Sea ECA.

The IMO Tier 3 NO_x limit is defined as follows:

NO_x [g/kWh] $= 9 \times rpm^{-0.2}$ when $130 < rpm < 2000$

The IMO Tier 3 NO_x emission level corresponds to an 80% reduction from the IMO Tier 2 NOx emission standard. The reduction can be reached by applying a secondary exhaust gas emission control system. A Selective Catalytic Reduction (SCR) system is an efficient way for diesel engines to reach the NOx reduction needed for the IMO Tier 3 standard.

If the Wärtsilä NOx Reducer SCR system is installed together with the engine, the engine + SCR installation complies with the maximum permissible NOx emission according to the IMO Tier 3 NOx emission standard and the Tier 3 EIAPP certificate will be delivered for the complete installation.

NOTE

The Dual Fuel engines fulfil the IMO Tier 3 NOx emission level as standard in gas mode operation without the need of a secondary exhaust gas emission control system.

13.2.1.1.2 Sulphur Oxides, SO_x emissions

Marpol Annex VI has set a maximum global fuel sulphur limit of currently 3,5% (from 1.1.2012) in weight for any fuel used on board a ship. Annex VI also contains provisions allowing for special SOx Emission Control Areas (SECA) to be established with more stringent controls on sulphur emissions. In a SECA, which currently comprises the Baltic Sea, the North Sea, the English Channel, the US Caribbean Sea and the area outside North America (200 nautical miles), the sulphur content of fuel oil used onboard a ship must currently not exceed 0,1 % in weight.

The Marpol Annex VI has undertaken a review with the intention to further reduce emissions from ships. The current and upcoming limits for fuel oil sulphur contents are presented in the following table.

Table 13-2 Fuel sulphur caps

Fuel sulphur cap	Area	Date of implementation
Max 3.5% S in fuel	Globally	1 January 2012
Max. 0.1% S in fuel	SECA Areas	1 January 2015
Max. 0.5% S in fuel	Globally	1 January 2020

Abatement technologies including scrubbers are allowed as alternatives to low sulphur fuels. The exhaust gas system can be applied to reduce the total emissions of sulphur oxides from ships, including both auxiliary and main propulsion engines, calculated as the total weight of sulphur dioxide emissions.

13.2.2 Other Legislations

There are also other local legislations in force in particular regions.

13.3 Methods to reduce exhaust emissions

All standard Wärtsilä engines meet the NOx emission level set by the IMO (International Maritime Organisation) and most of the local emission levels without any modifications. Wärtsilä has also developed solutions to significantly reduce NOx emissions when this is required.

Diesel engine exhaust emissions can be reduced either with primary or secondary methods. The primary methods limit the formation of specific emissions during the combustion process. The secondary methods reduce emission components after formation as they pass through the exhaust gas system.

For dual fuel engines same methods as mentioned above can be used to reduce exhaust emissions when running in diesel mode. In gas mode there is no need for scrubber or SCR.

Refer to the "*Wärtsilä Environmental Product Guide*" for information about exhaust gas emission control systems.

15. Foundation

Engines can be either rigidly mounted on chocks, or resiliently mounted on rubber elements. If resilient mounting is considered, Wärtsilä must be informed about existing excitations such as propeller blade passing frequency. Dynamic forces caused by the engine are listed in the chapter *Vibration and noise*.

15.1 Steel structure design

The system oil tank may not extend under the reduction gear, if the engine is of dry sump type and the oil tank is located beneath the engine foundation. Neither should the tank extend under the support bearing, in case there is a PTO arrangement in the free end. The oil tank must also be symmetrically located in transverse direction under the engine.

The foundation and the double bottom should be as stiff as possible in all directions to absorb the dynamic forces caused by the engine, reduction gear and thrust bearing. The foundation should be dimensioned and designed so that harmful deformations are avoided.

The foundation of the driven equipment must be integrated with the engine foundation.

15.2 Mounting of main engines

15.2.1 Rigid mounting

Main engines can be rigidly mounted to the foundation either on steel chocks or resin chocks.

The holding down bolts are through-bolts with a lock nut at the lower end and a hydraulically tightened nut at the upper end. The tool included in the standard set of engine tools is used for hydraulic tightening of the holding down bolts. Two of the holding down bolts are fitted bolts and the rest are clearance bolts. The two Ø43H7/n6 fitted bolts are located closest to the flywheel, one on each side of the engine.

A distance sleeve should be used together with the fitted bolts. The distance sleeve must be mounted between the seating top plate and the lower nut in order to provide a sufficient guiding length for the fitted bolt in the seating top plate. The guiding length in the seating top plate should be at least equal to the bolt diameter.

The design of the holding down bolts is shown in the foundation drawings. It is recommended that the bolts are made from a high-strength steel, e.g. 42CrMo4 or similar. A high strength material makes it possible to use a higher bolt tension, which results in a larger bolt elongation (strain). A large bolt elongation improves the safety against loosening of the nuts.

To avoid sticking during installation and gradual reduction of tightening tension due to unevenness in threads, the threads should be machined to a finer tolerance than normal threads. The bolt thread must fulfil tolerance 6g and the nut thread must fulfil tolerance 6H. In order to avoid bending stress in the bolts and to ensure proper fastening, the contact face of the nut underneath the seating top plate should be counterbored.

Lateral supports must be installed for all engines. One pair of supports should be located at flywheel end and one pair (at least) near the middle of the engine. The lateral supports are to be welded to the seating top plate before fitting the chocks. The wedges in the supports are to be installed without clearance, when the engine has reached normal operating temperature. The wedges are then to be secured in position with welds. An acceptable contact surface must be obtained on the wedges of the supports.

15.2.1.1 Resin chocks

The recommended dimensions of resin chocks are 150 x 400 mm. The total surface pressure on the resin must not exceed the maximum permissible value, which is determined by the type of resin and the requirements of the classification society. It is recommended to select a resin type that is approved by the relevant classification society for a total surface pressure of 5 N/mm^2. (A typical conservative value is Ptot 3.5 N/mm^2).

During normal conditions, the support face of the engine feet has a maximum temperature of about 75°C, which should be considered when selecting the type of resin.

The bolts must be made as tensile bolts with a reduced shank diameter to ensure a sufficient elongation since the bolt force is limited by the permissible surface pressure on the resin. For a given bolt diameter the permissible bolt tension is limited either by the strength of the bolt material (max. stress 80% of the yield strength), or by the maximum permissible surface pressure on the resin.

15.2.1.2 Steel chocks

The top plates of the foundation girders are to be inclined outwards with regard to the centre line of the engine. The inclination of the supporting surface should be 1/100 and it should be machined so that a contact surface of at least 75% is obtained against the chocks.

Recommended chock dimensions are 250 x 200 mm and the chocks must have an inclination of 1:100, inwards with regard to the engine centre line. The cut-out in the chocks for the clearance bolts shall be 44 mm (M42 bolts), while the hole in the chocks for the fitted bolts shall be drilled and reamed to the correct size (Ø43H7) when the engine is finally aligned to the reduction gear.

The design of the holding down bolts is shown the foundation drawings. The bolts are designed as tensile bolts with a reduced shank diameter to achieve a large elongation, which improves the safety against loosening of the nuts.

15.2.1.3 Steel chocks with adjustable height

As an alternative to resin chocks or conventional steel chocks it is also permitted to install the engine on adjustable steel chocks. The chock height is adjustable between 45 mm and 65 mm for the approved type of chock. There must be a chock of adequate size at the position of each holding down bolt.

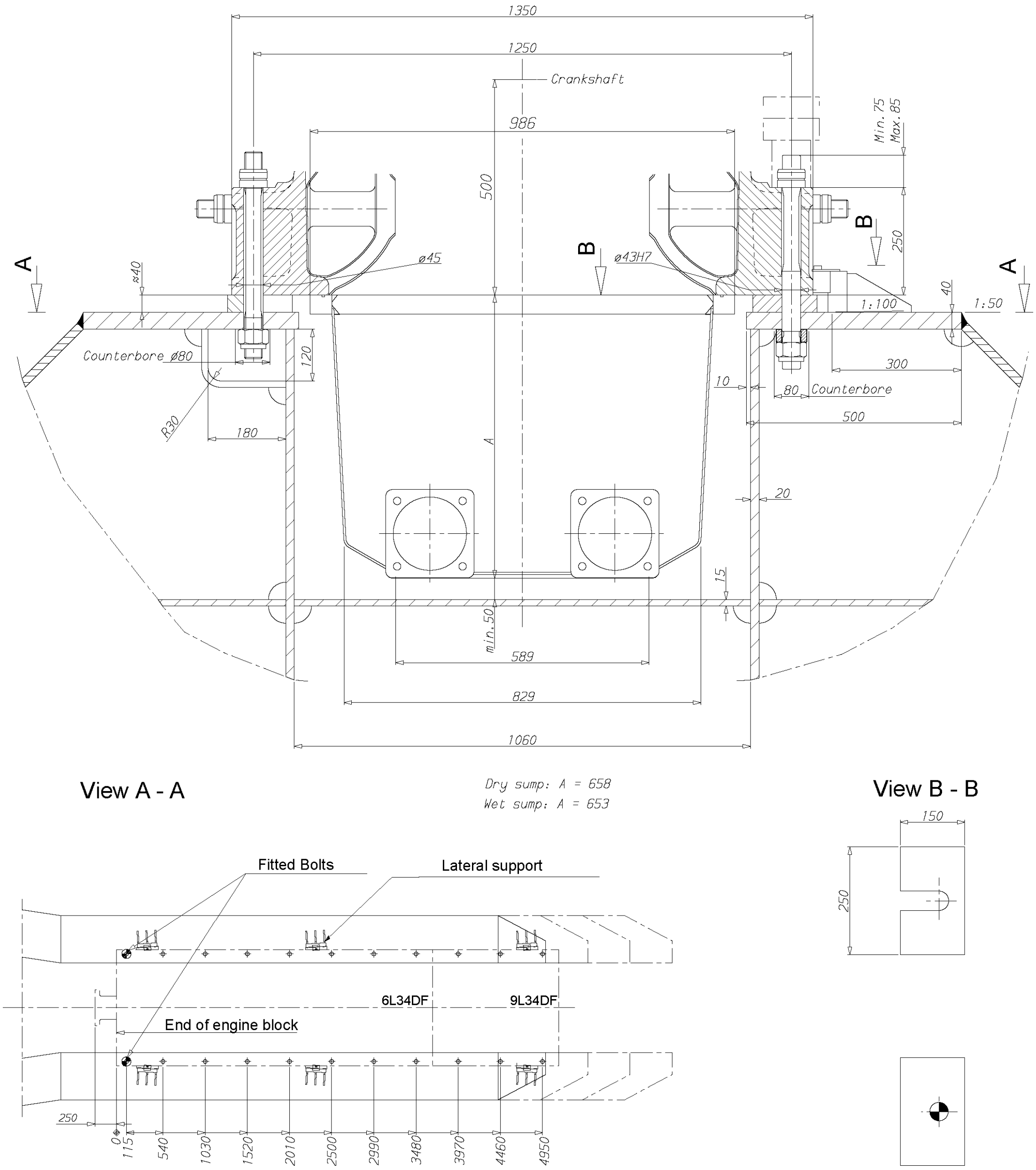
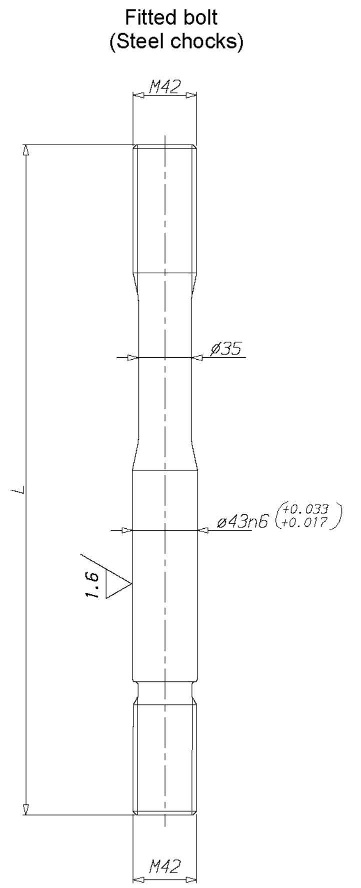
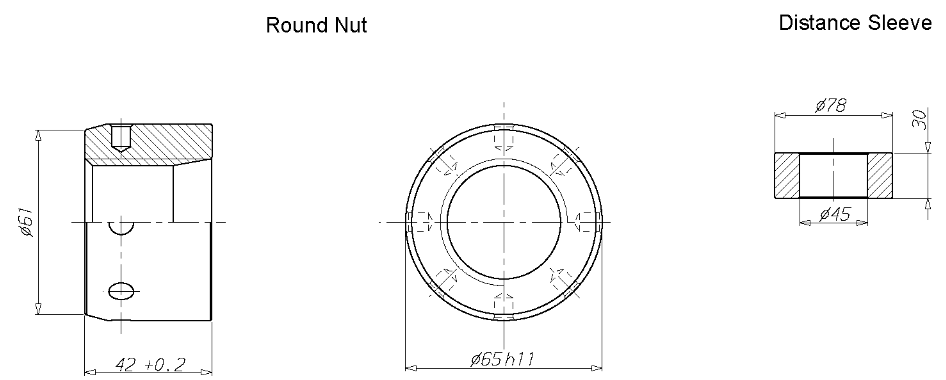


Fig 15-1 **Main engine seating and fastening, in-line engines, steel chocks (DAAE085777)**





Number of pieces per engine			
	W 6L34DF	**W 8L34DF**	**W 9L34DF**
Fitted bolt	2	2	2
Clearance bolt	14	18	20
Round nut	16	20	22
Lock nut	16	20	22
Distance sleeve	2	2	2
Lateral support	4	4	6
Chocks	16	20	22

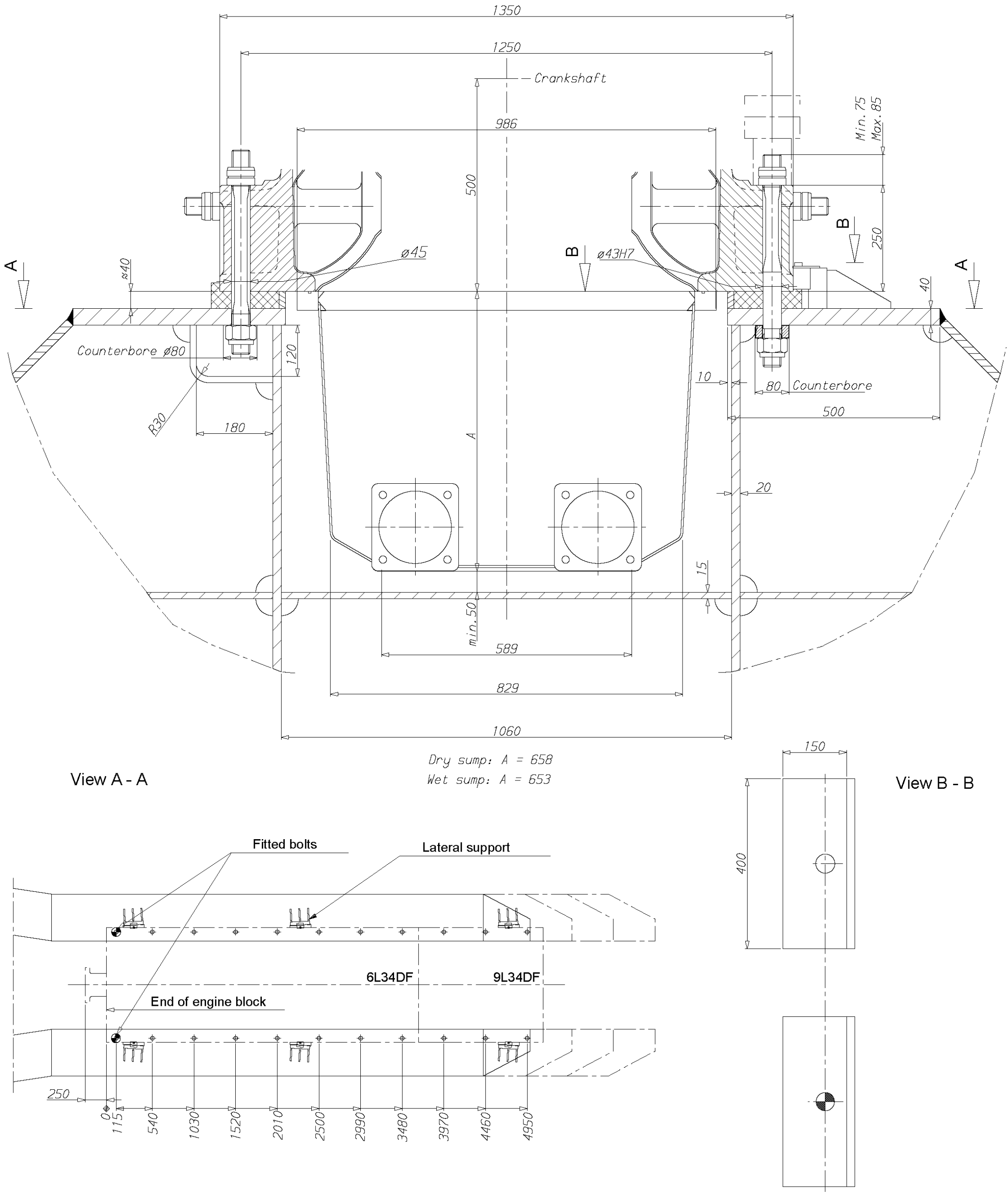
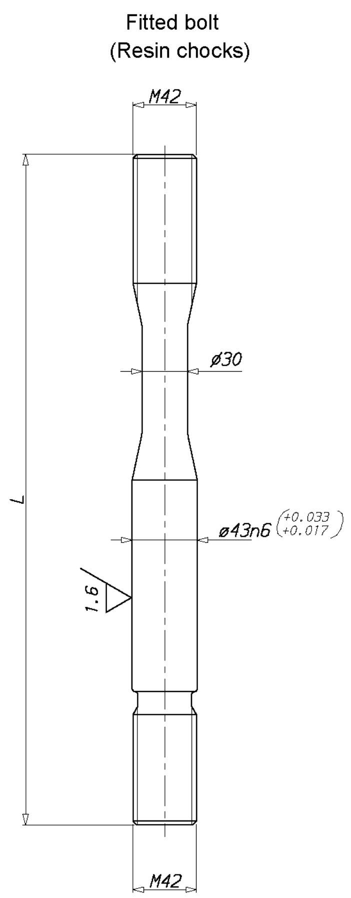
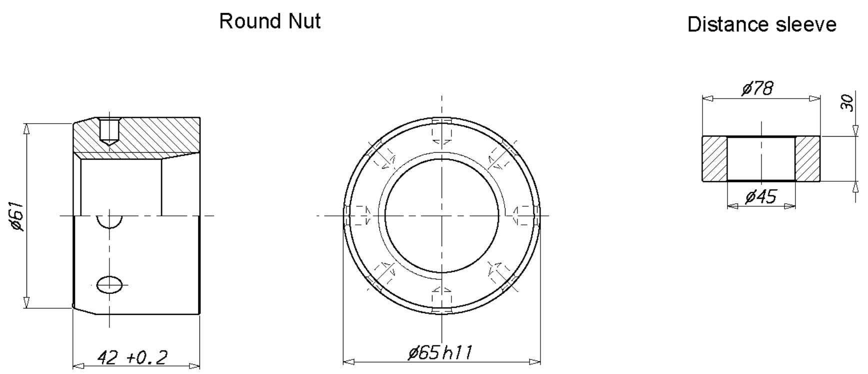


Fig 15-2 **Main engine seating and fastening, in-line engines, resin chocks (DAAE085778)**





Number of pieces per engine			
	W 6L34DF	W 8L34DF	W 9L34DF
Fitted bolt	2	2	2
Clearance bolt	14	18	20
Round nut	16	20	22
Lock nut	16	20	22
Distance sleeve	2	2	2
Lateral support	4	4	6
Chocks	16	20	22

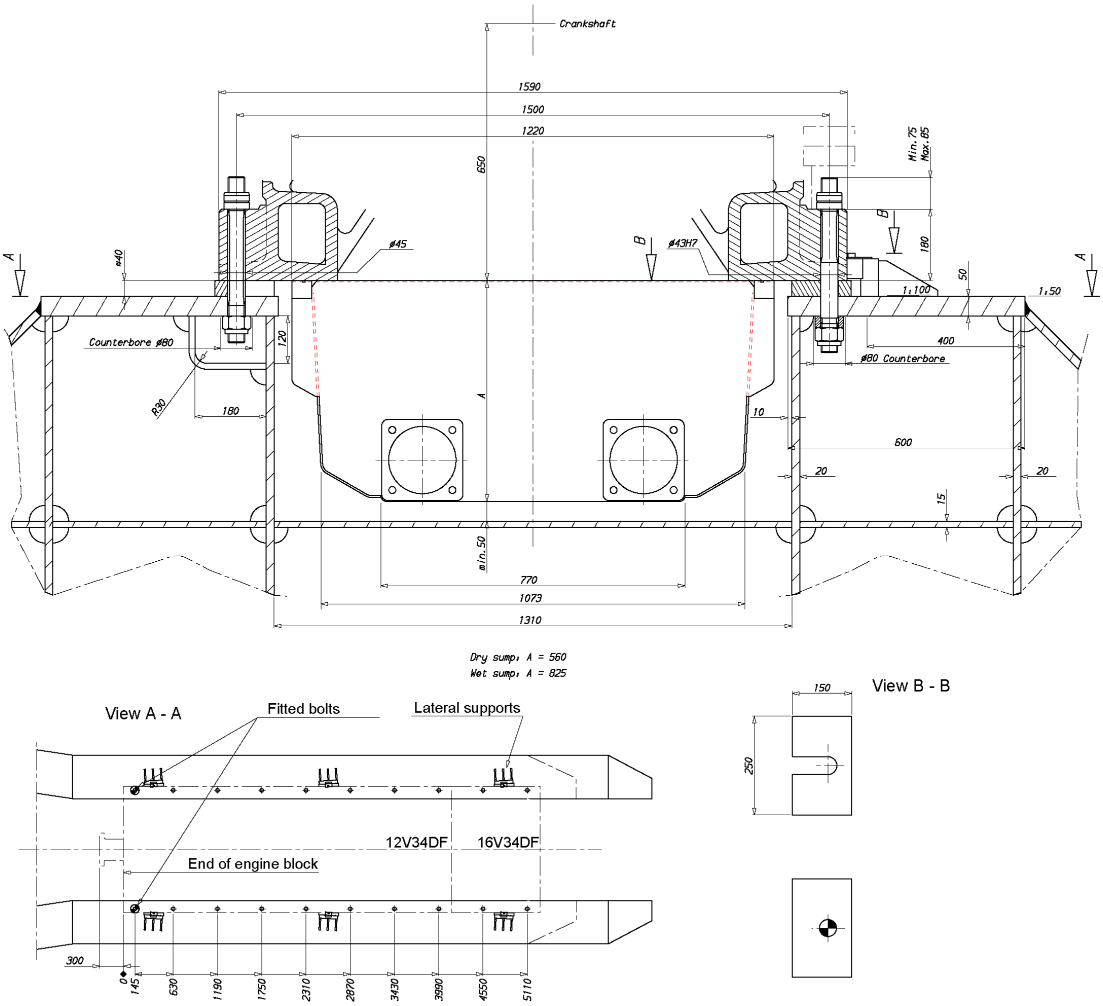
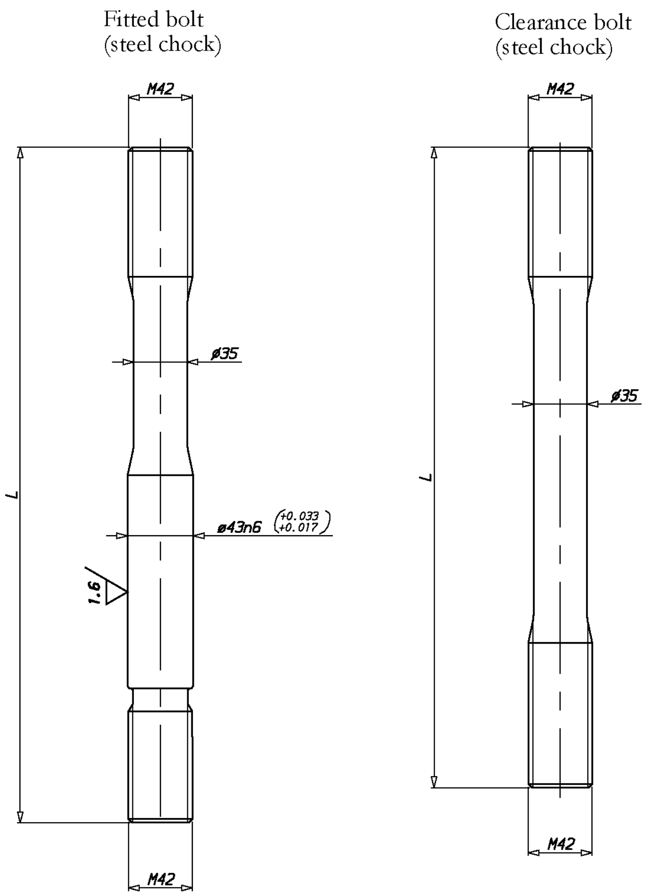
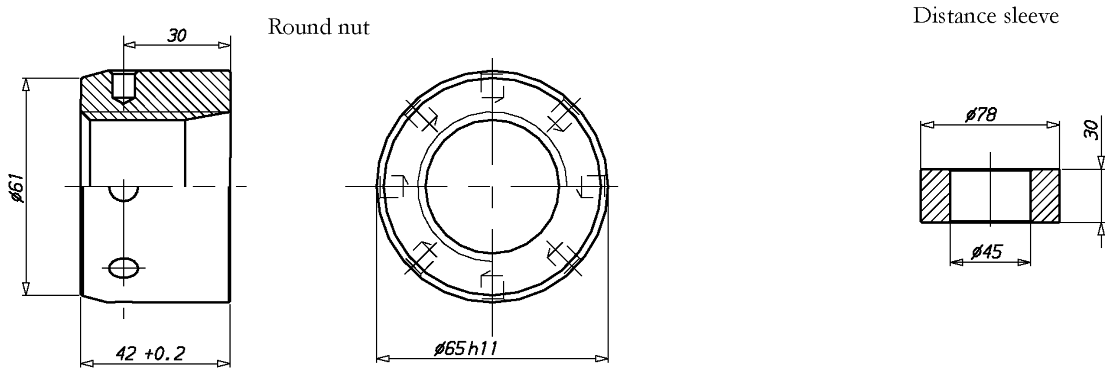


Fig 15-3 Main engine seating and fastening, V-engines, steel chocks (DAAE085776)





Number of pieces per engine		
	W 12V34DF	W 16V34DF
Fitted bolt	2	2
Clearance bolt	14	18
Round nut	16	20
Lock nut	16	20
Distance sleeve	2	2
Lateral support	4	6
Chocks	16	20

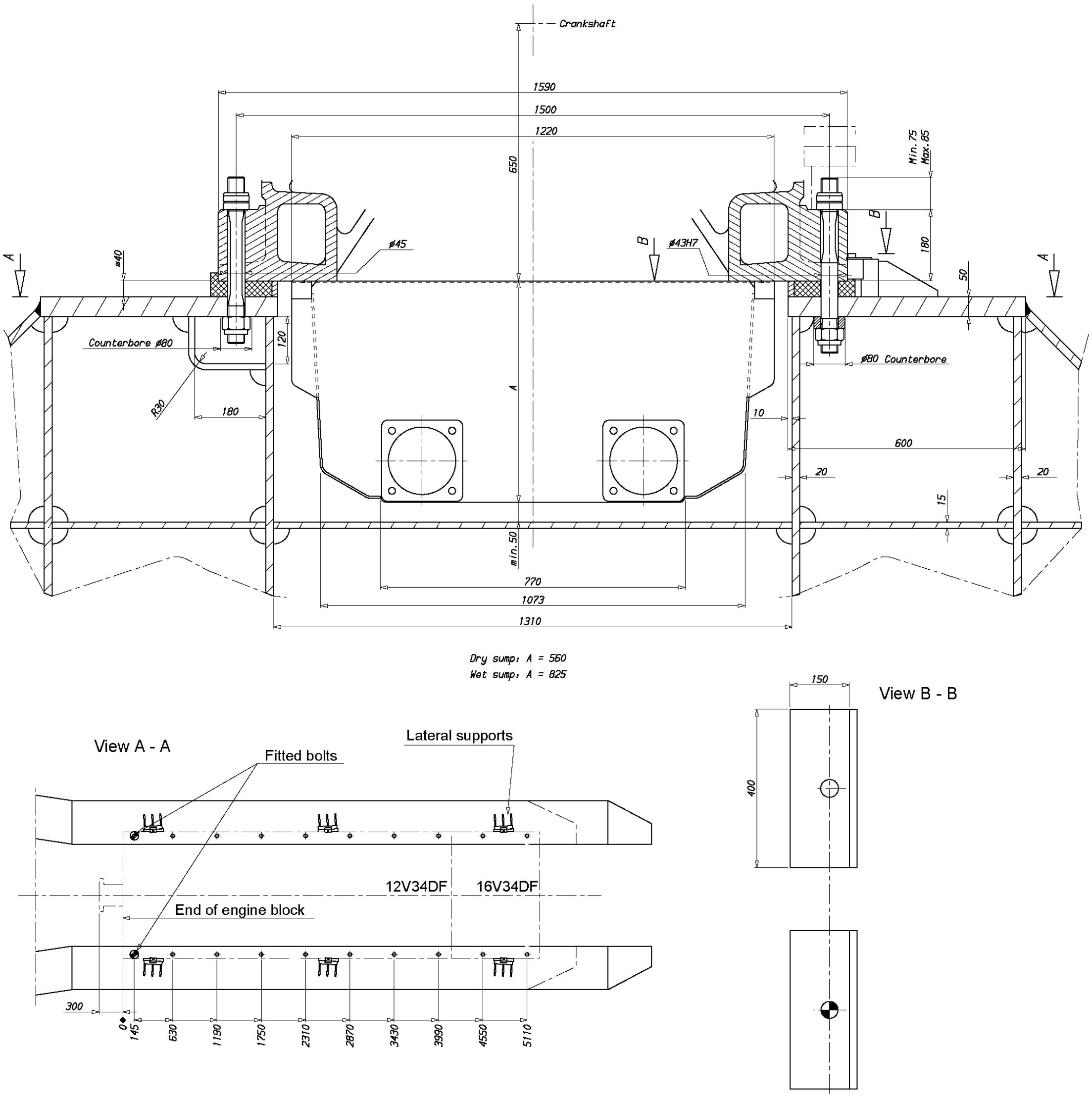
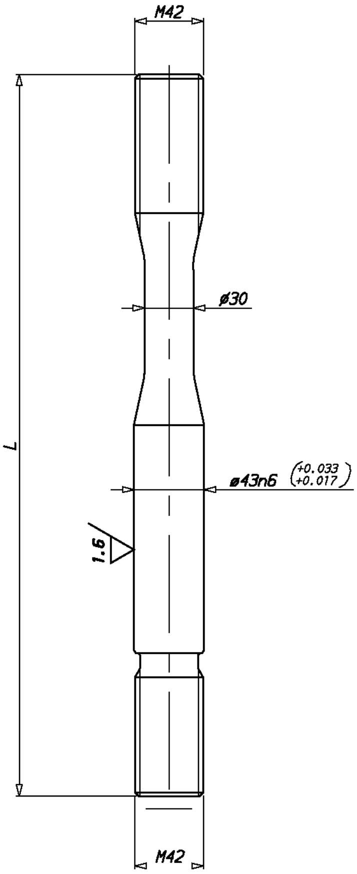
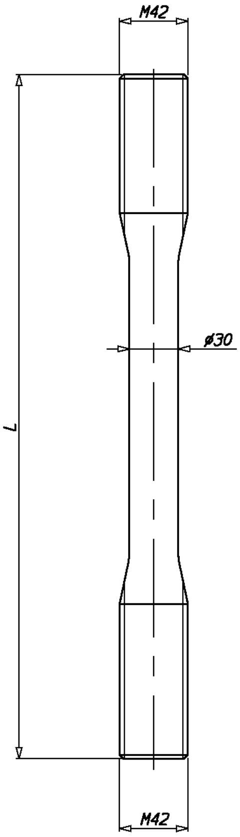


Fig 15-4 Main engine seating and fastening, V-engines, resin chocks (DAAE085781)

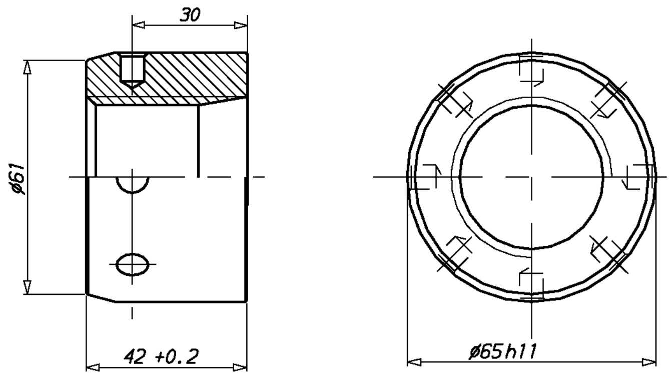
Fitted bolt
(resin chock)



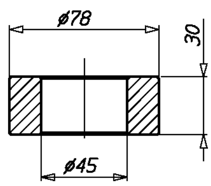
Clearance bolt
(resin chock)



Round nut



Distance sleeve



Number of pieces per engine		
	W 12V34DF	W 16V34DF
Fitted bolt	2	2
Clearance bolt	14	18
Round nut	16	20
Lock nut	16	20
Distance sleeve	2	2
Lateral support	4	6
Chocks	16	20

15.2.2 Resilient mounting

In order to reduce vibrations and structure borne noise, main engines can be resiliently mounted on rubber elements. The transmission of forces emitted by the engine is 10-20% when using resilient mounting.

Two different mounting arrangements are applied. Cylinder configurations 6L, 8L, 12V and 16V are mounted on conical rubber mounts, which are similar to the mounts used under generating sets. The mounts are fastened directly to the engine feet with a hydraulically tightened bolt. To enable drilling of holes in the foundation after final alignment adjustments the mount is fastened to an intermediate steel plate, which is fixed to the foundation with one bolt. The hole in the foundation for this bolt can be drilled through the engine foot. A resin chock is cast under the intermediate steel plate.

Cylinder configuration 9L is mounted on cylindrical rubber elements. These rubber elements are mounted to steel plates in groups, forming eight units. These units, or resilient elements, each consist of an upper steel plate that is fastened directly to the engine feet, rubber elements and a lower steel plate that is fastened to the foundation. The holes in the foundation for the fastening bolts can be drilled through the holes in the engine feet, when the engine is finally aligned to the reduction gear. The resilient elements are compressed to the calculated height under load by using M30 bolts through the engine feet and distance pieces between the two steel plates. Resin chocks are then cast under the resilient elements. Shims are provided for installation between the engine feet and the resilient elements to facilitate alignment adjustments in vertical direction. Steel chocks must be used under the side and end buffers located at each corner if the engine.

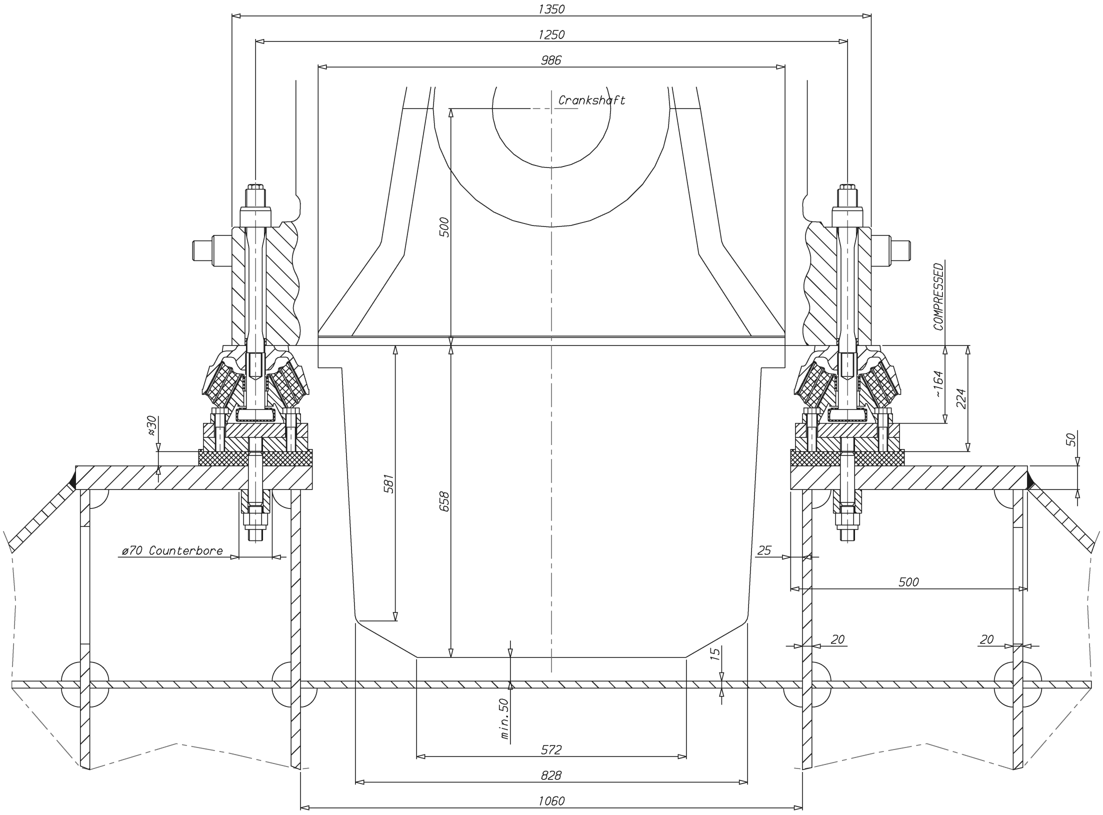


Fig 15-5 **Principle of resilient mounting, W6L34DF and W8L34DF (DAAE048811)**

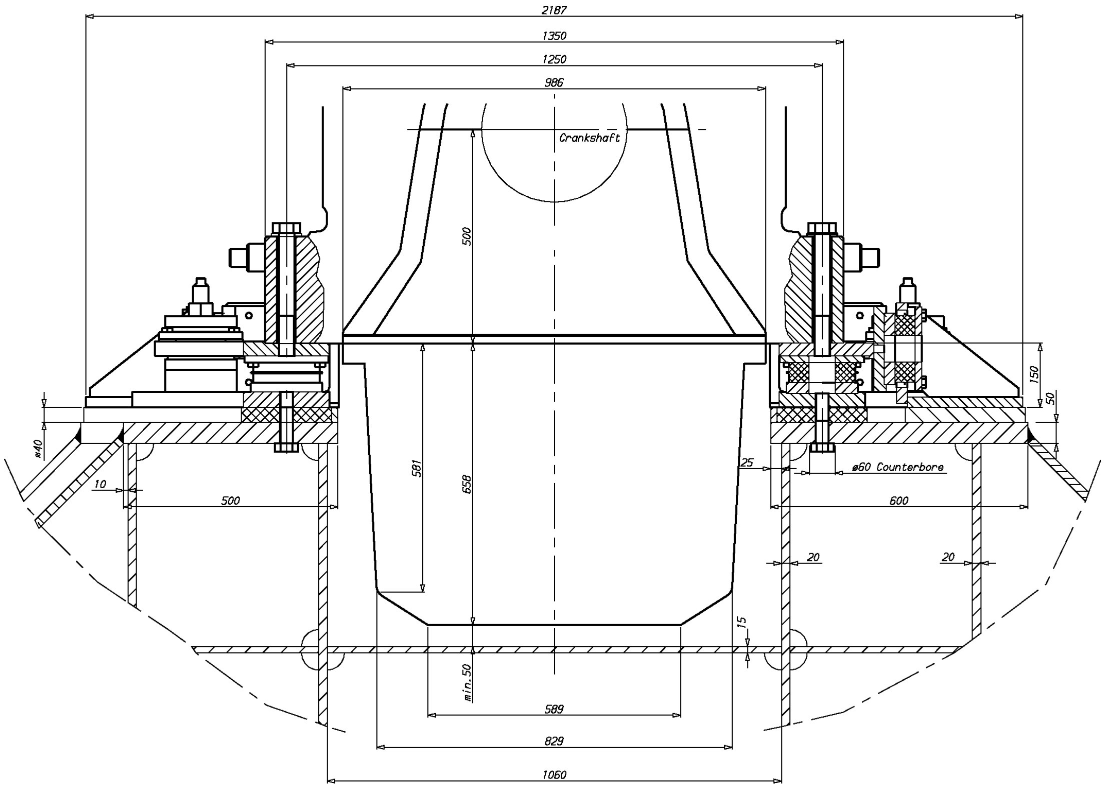


Fig 15-6 Principle of resilient mounting, W9L34DF (V69A0247A)

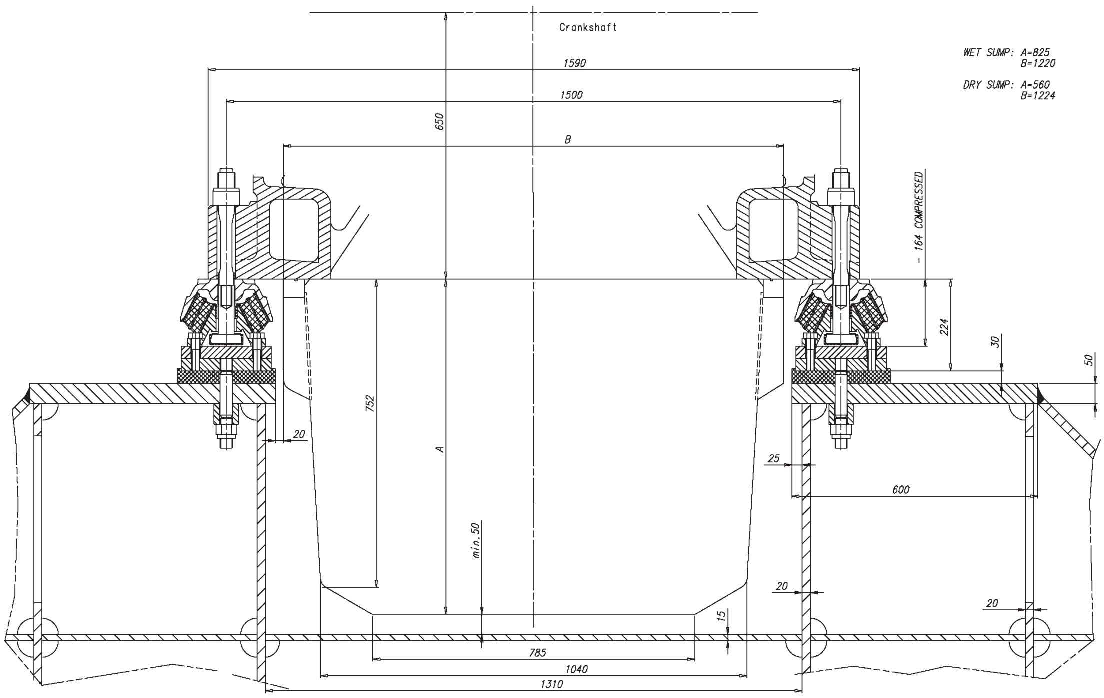


Fig 15-7 Principle of resilient mounting, W12V34DF and W16V34DF (DAAE041111A)

15.3 Mounting of generating sets

15.3.1 Generator feet design

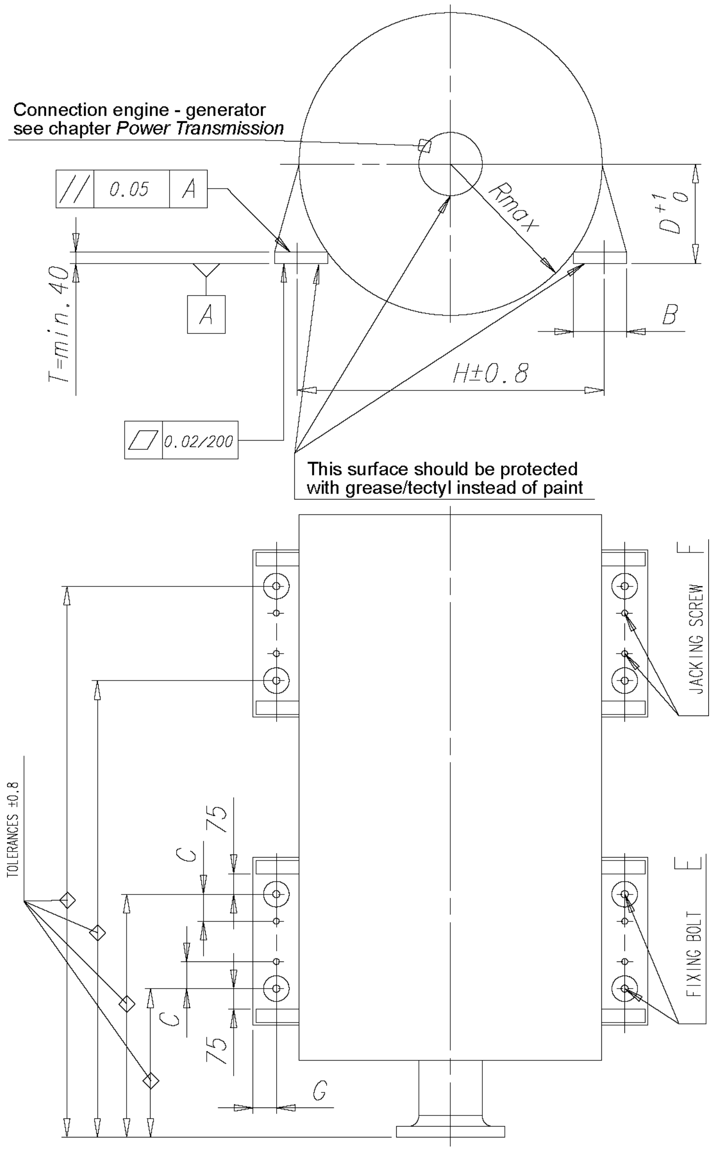


Fig 15-8 **Distance between fixing bolts on generator (DAAE084469)**

H [mm]	W 6L34DF Rmax [mm]	W 8L34DF Rmax [mm]	W 9L34DF Rmax [mm]	W 12V34DF Rmax [mm]	W 16V34DF Rmax [mm]
1400	715	-	-	-	-
1600	810	810	810	-	-
1800	-	905	905	985	985
1950	-	980	980	1045	1045
2200	-	-	1090	-	-

Engine	G [mm]	F	E [mm]	D [mm]	C [mm]	B [mm]
W L34DF	85	M24 or M27	Ø35	475	100	170
W V34DF	100	M30	Ø48	615	130	200

15.3.2 Resilient mounting

Generating sets, comprising engine and generator mounted on a common base frame, are usually installed on resilient mounts on the foundation in the ship.

The resilient mounts reduce the structure borne noise transmitted to the ship and also serve to protect the generating set bearings from possible fretting caused by hull vibration.

The number of mounts and their location is calculated to avoid resonance with excitations from the generating set engine, the main engine and the propeller.

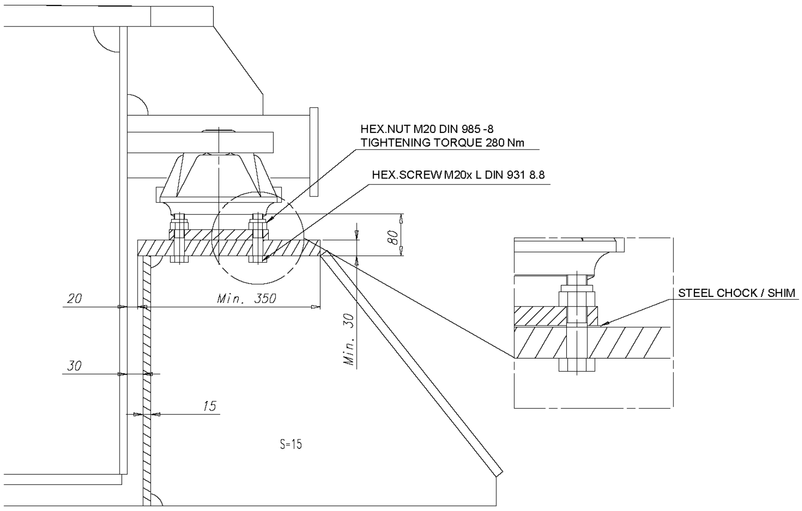
NOTE

To avoid induced oscillation of the generating set, the following data must be sent by the shipyard to Wärtsilä at the design stage:

- main engine speed [RPM] and number of cylinders
- propeller shaft speed [RPM] and number of propeller blades

The selected number of mounts and their final position is shown in the generating set drawing.

In-line engines



V-engines

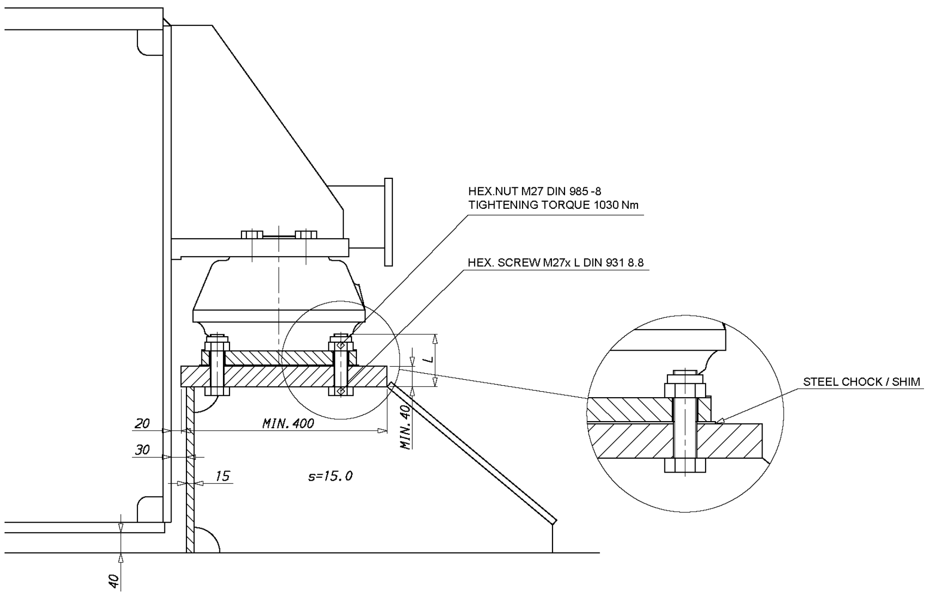


Fig 15-9 Recommended design of the generating set seating (3V46L0295d, DAAE020067a)

15.3.2.1 Rubber mounts

The generating set is mounted on conical resilient mounts, which are designed to withstand both compression and shear loads. In addition the mounts are equipped with an internal buffer to limit the movements of the generating set due to ship motions. Hence, no additional side or end buffers are required.

The rubber in the mounts is natural rubber and it must therefore be protected from oil, oily water and fuel.

The mounts should be evenly loaded, when the generating set is resting on the mounts. The maximum permissible variation in compression between mounts is 2.0 mm. If necessary, chocks or shims should be used to compensate for local tolerances. Only one shim is permitted under each mount.

The transmission of forces emitted by the engine is 10 -20% when using conical mounts. For the foundation design, see drawing 3V46L0295 (in-line engines) and 3V46L0294 (V-engines).

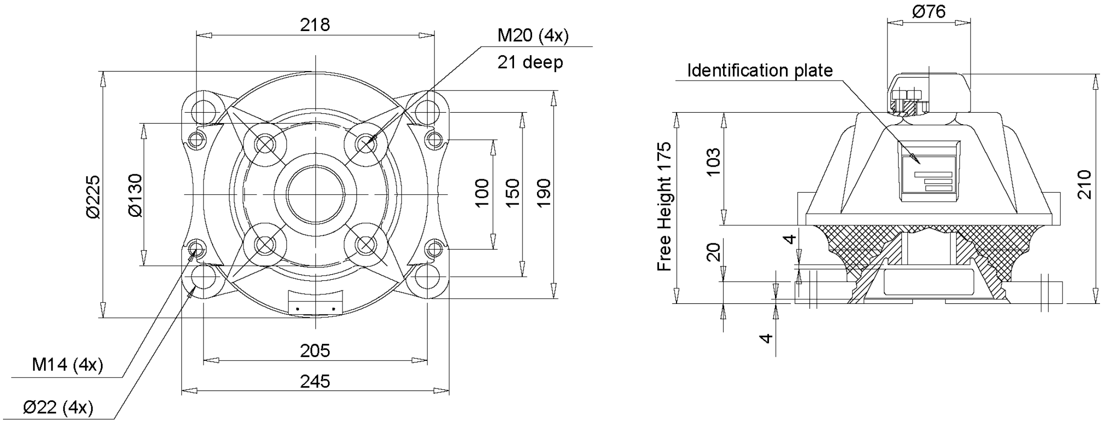


Fig 15-10 **Rubber mount, In-line engines (DAAE004230c)**

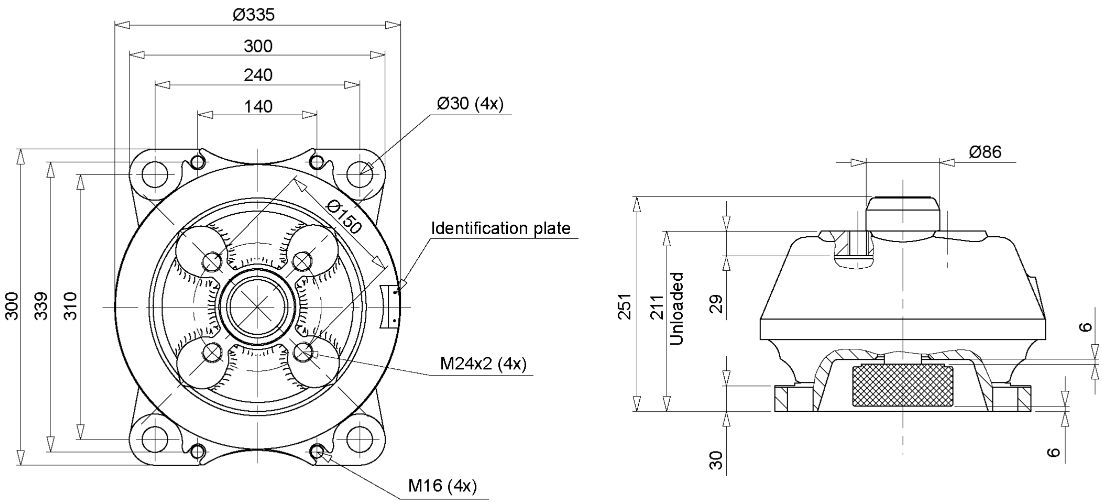


Fig 15-11 **Rubber mount, V-engines (DAAE018766b)**

15.4 Flexible pipe connections

When the engine or generating set is resiliently installed, all connections must be flexible and no grating nor ladders may be fixed to the engine or generating set. When installing the flexible pipe connections, unnecessary bending or stretching should be avoided. The external pipe must be precisely aligned to the fitting or flange on the engine. It is very important that the pipe clamps for the pipe outside the flexible connection must be very rigid and welded to the steel structure of the foundation to prevent vibrations, which could damage the flexible connection.

18. Engine Room Layout

18.1 Crankshaft distances

Minimum crankshaft distances have to be followed in order to provide sufficient space between engines for maintenance and operation.

18.1.1 Main engines

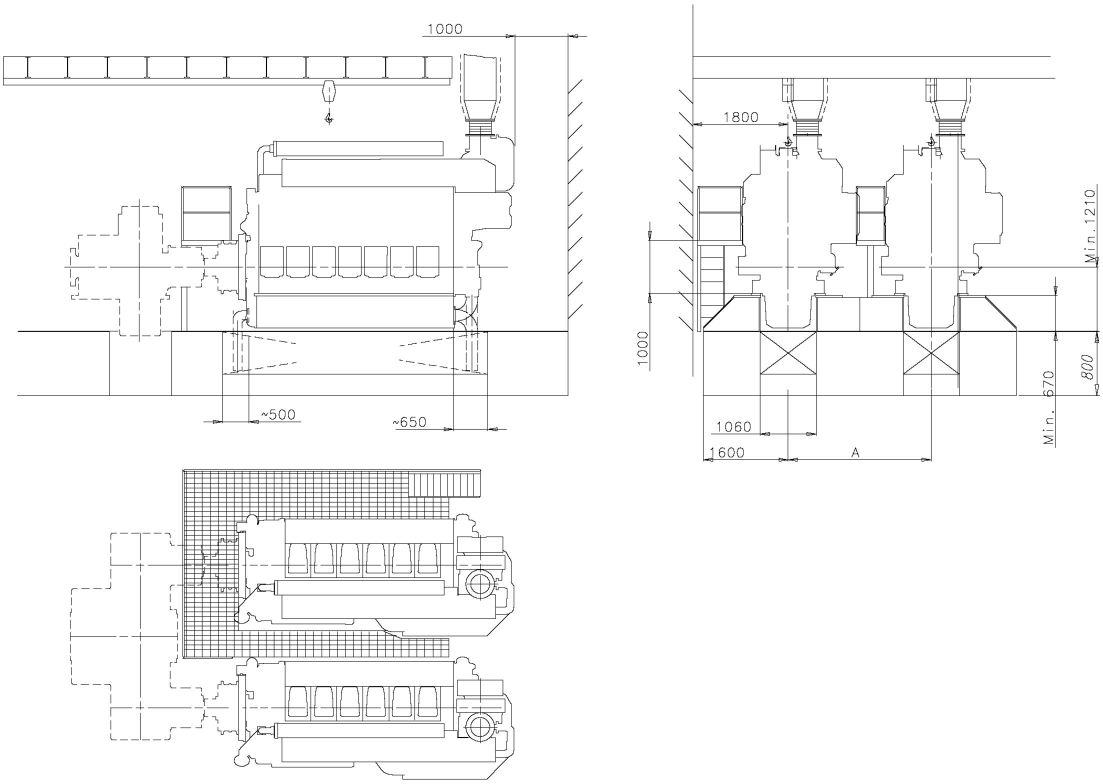


Fig 18-1 Crankshaft distances, in-line engines (DAAE082974B)

Engine type	A [mm]
W 6L34DF	2700
W 8L34DF	2700
W 9L34DF	2700

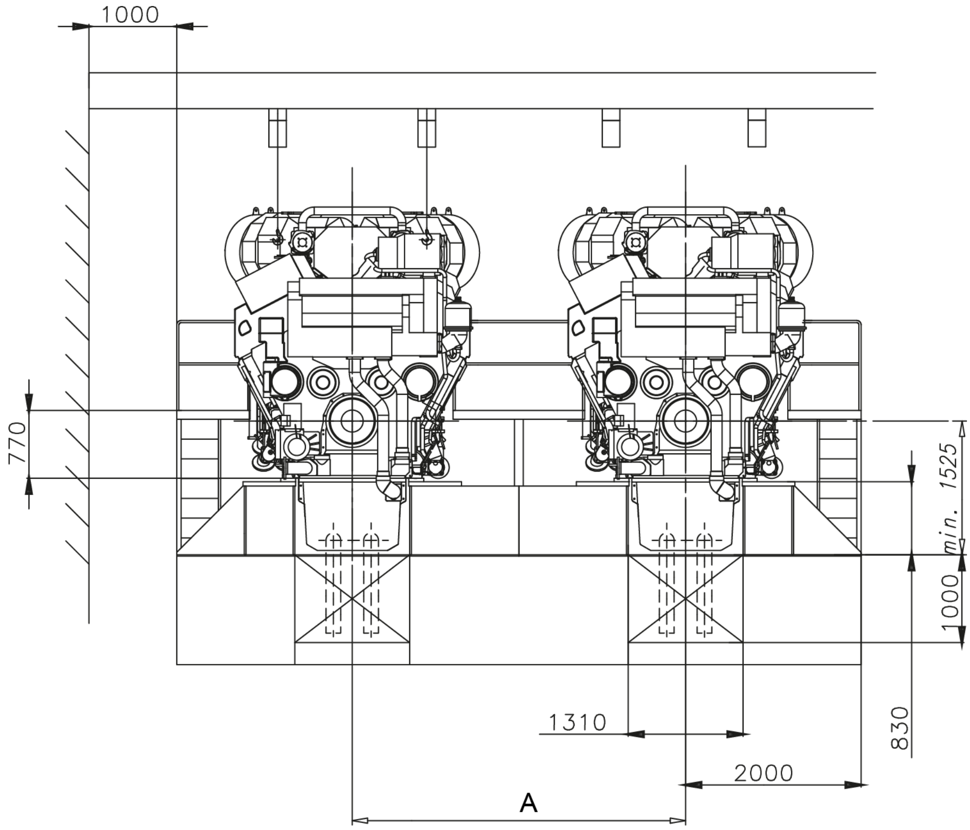


Fig 18-2 Crankshaft distances, V-engines (DAAF073294)

Engine type	A [mm]
TC with air filter/silencer on turbocharger	3700
Air duct connected to TC	3800

18.1.2 Generating sets

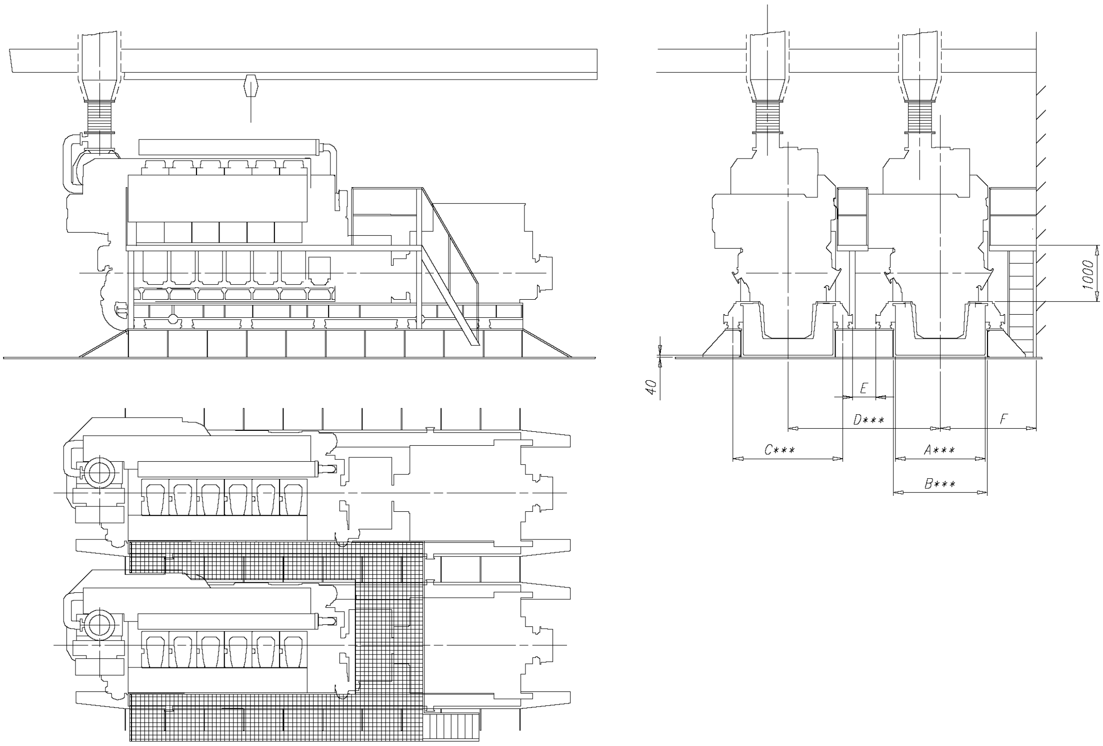


Fig 18-3 Crankshaft distances, in-line engines (DAAE082973A)

Engine type	A ***	B ***	C ***	D ***	E	F
W 6L34DF	1800	1860	2110	2800	310	1700
W 8L34DF	1800	1860	2110	2800	310	1700
W 9L34DF	2000	2060	2310	2800	210	2000

All dimensions in mm.

*** Dependent on the generator type.

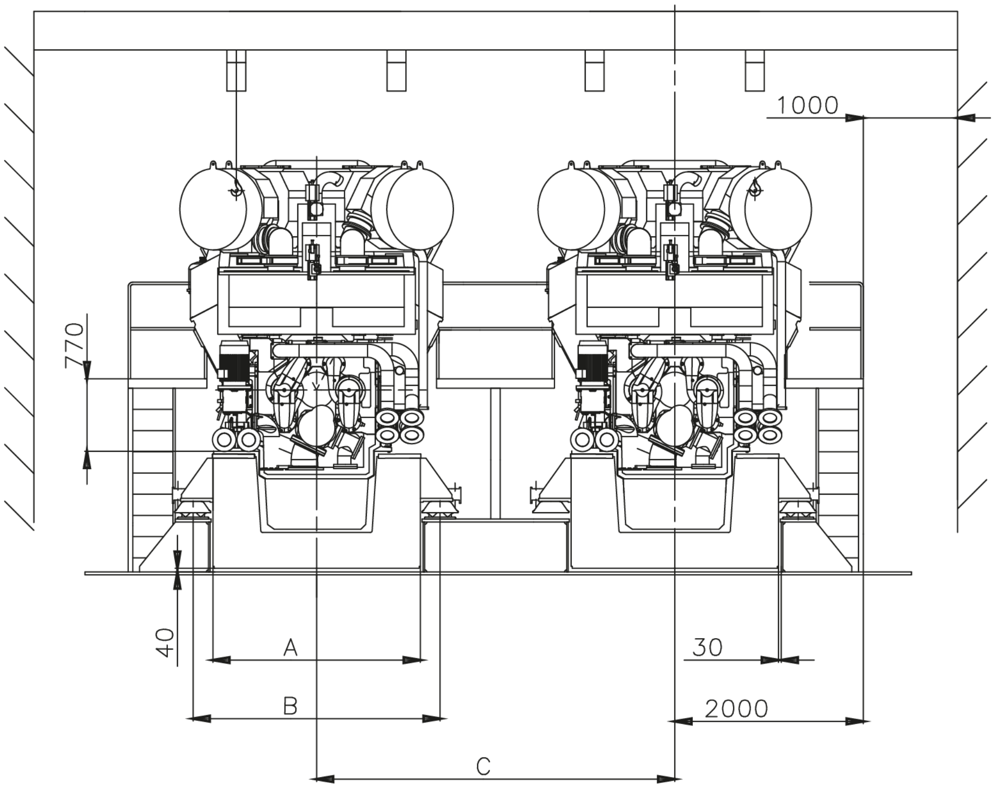


Fig 18-4 Crankshaft distances, V-engines (DAAF073293)

Engine type	A	B	C
W 12V34DF	2200	2620	min. 3800
W 16V34DF	2200	2620	min. 3800

All dimensions in mm.

18.1.3 Father-and-son arrangement

When connecting two engines of different type and/or size to the same reduction gear the minimum crankshaft distance has to be evaluated case by case. However, some general guidelines can be given:

- It is essential to check that all engine components can be dismounted. The most critical are usually turbochargers and charge air coolers.
- When using a combination of in-line and v-engine, the operating side of in-line engine should face the v-engine in order to minimize the distance between crankshafts.
- Special care has to be taken checking the maintenance platform elevation between the engines to avoid structures that obstruct maintenance.

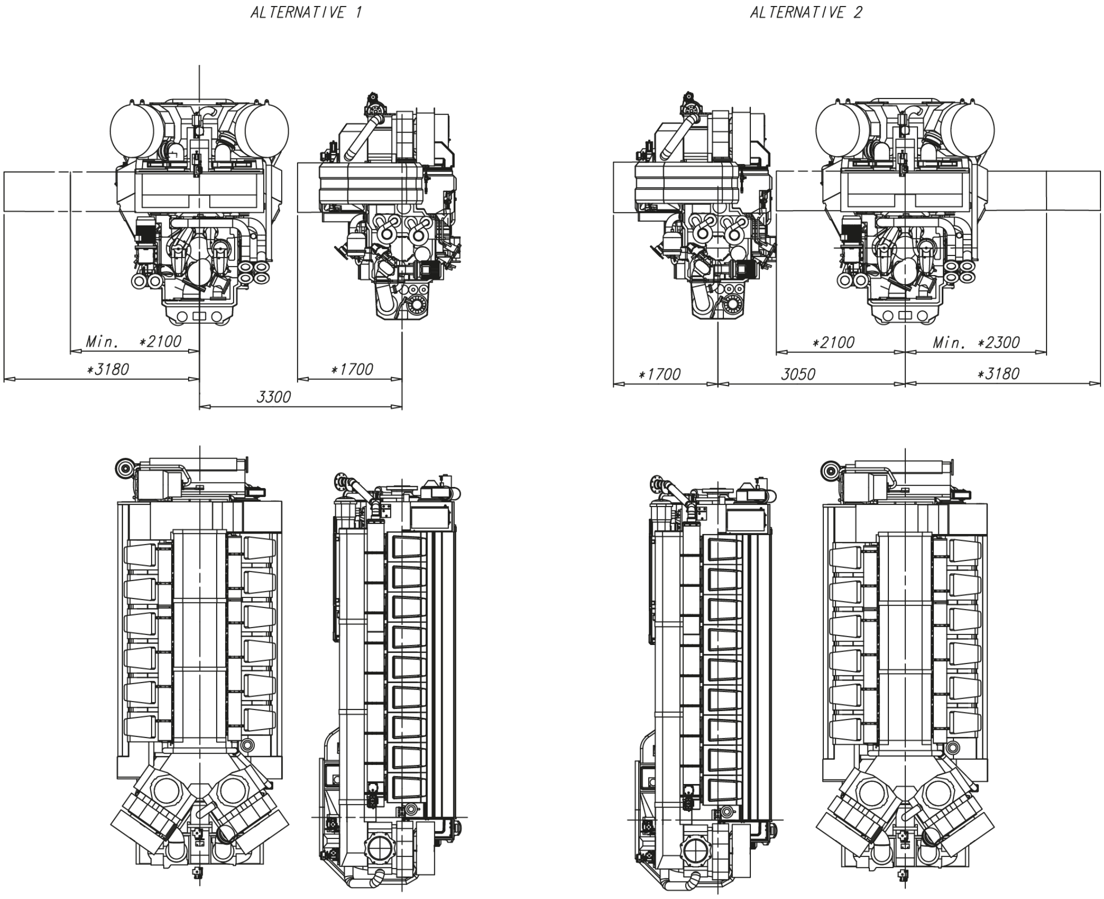


Fig 18-5 **Example of father-and-son arrangement, TC in free end (DAAF073307)**

All dimensions in mm. *) 50mm for clearance included.

18.1.4 Distance from adjacent intermediate/propeller shaft

Some machinery arrangements feature an intermediate shaft or propeller shaft running adjacent to engine. To allow adequate space for engine inspections and maintenance there has to be sufficient free space between the intermediate/propeller shaft and the engine. To enable safe working conditions the shaft has to be covered. It must be noticed that also dimensions of this cover have to be taken into account when determining the shaft distances in order to fulfil the requirement for minimum free space between the shaft and the engine.

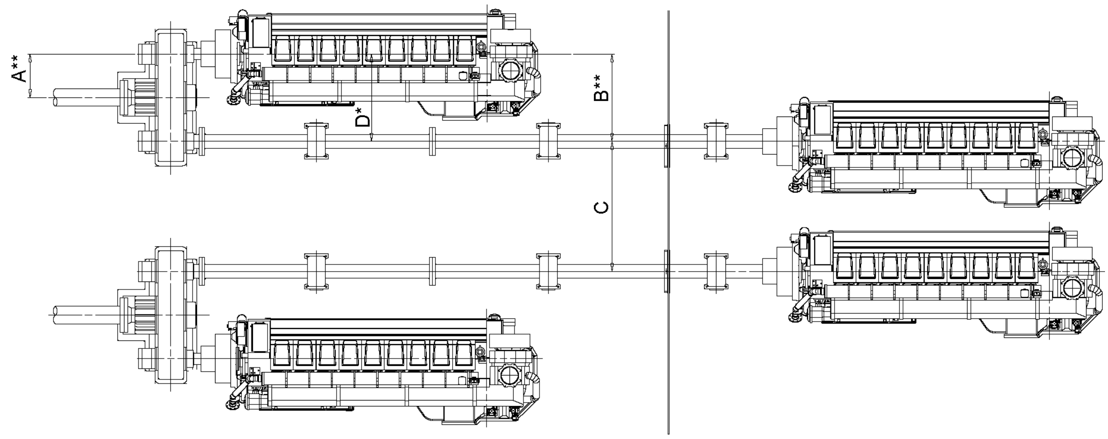


Fig 18-6 Main engine arrangement, in-line engines (DAAE086973B)

Engine type	A**	B**	C	D*
W 6L34DF	940	1880	2700	1480
W 8L34DF	940	1880	2700	1480
W 9L34DF	940	1880	2700	1480

Notes:
All dimensions in mm.
Intermediate shaft diameter to be determined case by case
* Depending on type of shaft bearing
** Depends on the type of gearbox

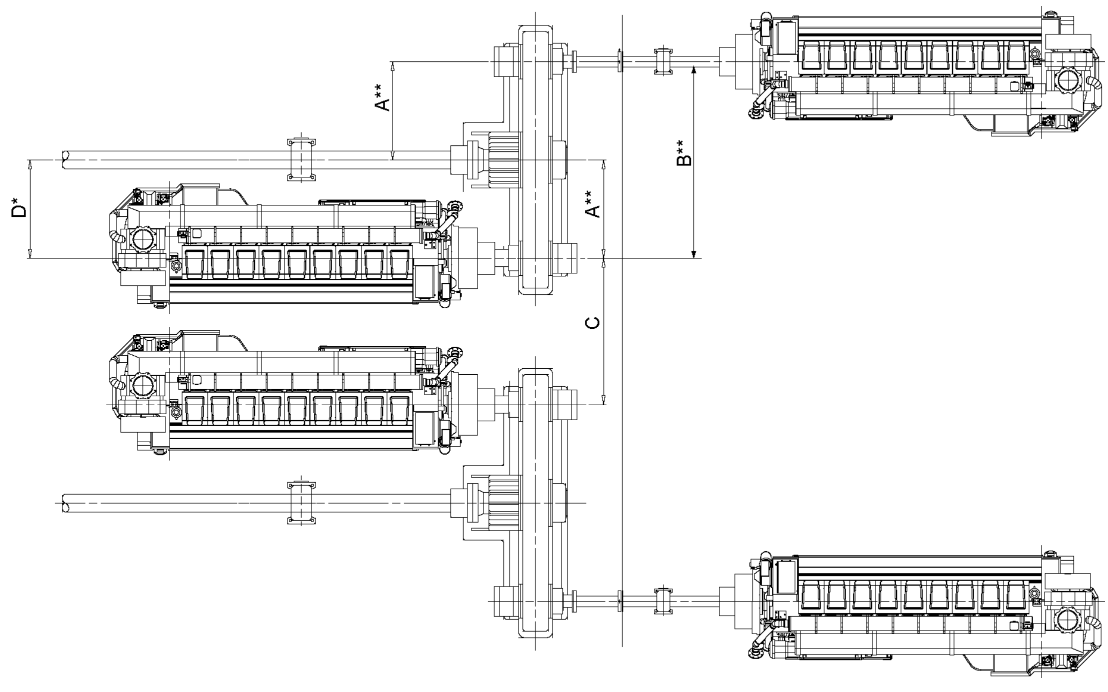


Fig 18-7 Main engine arrangement, in-line engines (DAAE086972B)

Engine type	A**	B**	C	D*
W 6L34DF	1880	3760	2700	1480
W 8L34DF	1880	3760	2700	1480
W 9L34DF	1880	3760	2700	1480

Notes:
All dimensions in mm.
Intermediate shaft diameter to be determined case by case.
* Depends on type of shaft bearing
** Depends on the type of gearbox

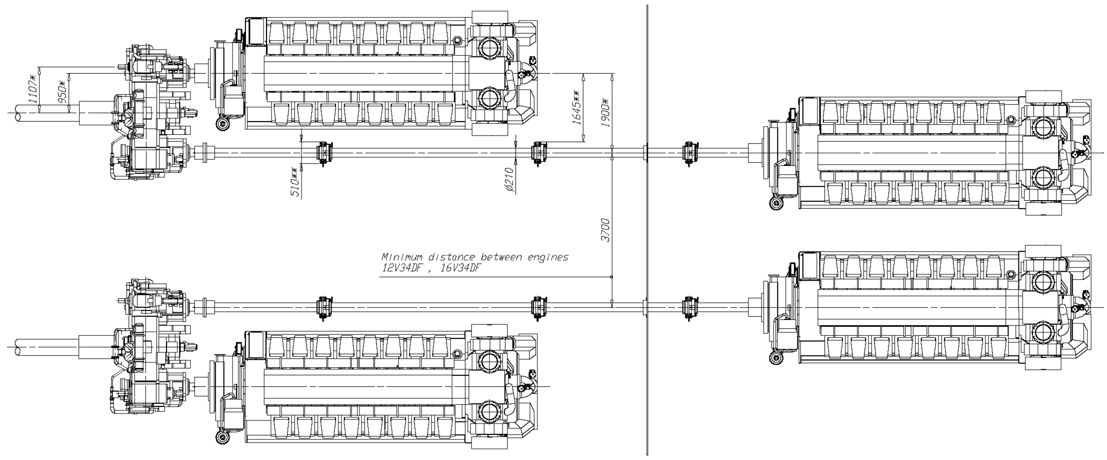


Fig 18-8 Main engine arrangement, V-engines (DAAE083977, DAAF068349)

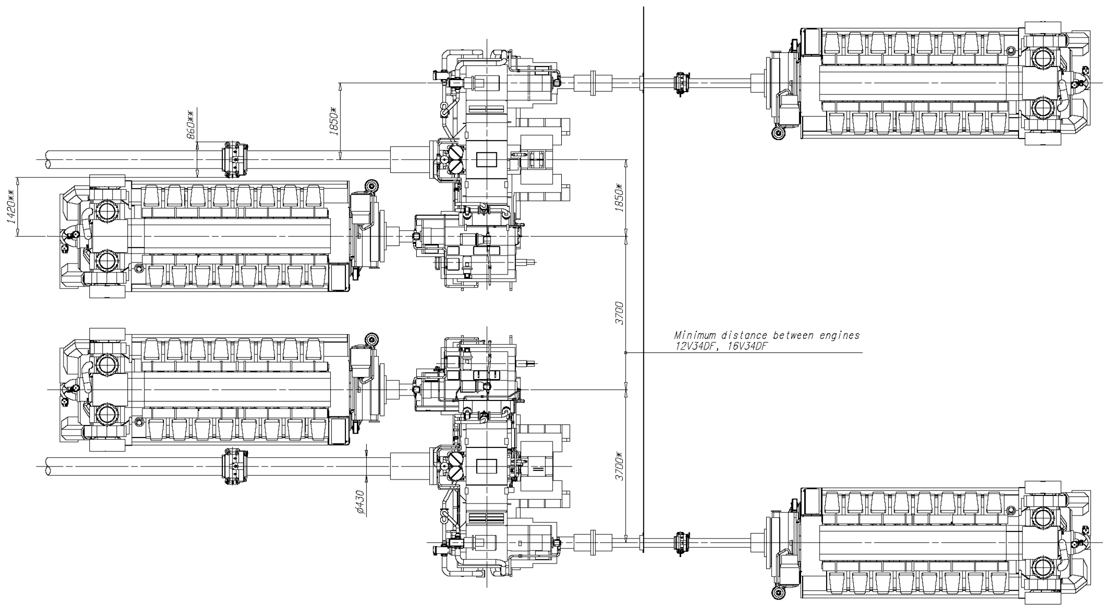


Fig 18-9 Main engine arrangement, V-engines (DAAE083975, DAAF068345)

Notes:
All dimensions in mm.
Intermediate shaft diameter to be determined case by case
* Depends on type of gearbox
** Depends on type of shaft bearing

18.2 Space requirements for maintenance

18.2.1 Working space around the engine

The required working space around the engine is mainly determined by the dismounting dimensions of engine components, and space requirement of some special tools. It is especially important that no obstructive structures are built next to engine driven pumps, as well as camshaft and crankcase doors.

However, also at locations where no space is required for dismounting of engine parts, a minimum of 1000 mm free space is recommended for maintenance operations everywhere around the engine.

18.2.2 Engine room height and lifting equipment

The required engine room height is determined by the transportation routes for engine parts. If there is sufficient space in transverse and longitudinal direction, there is no need to transport engine parts over the rocker arm covers or over the exhaust pipe and in such case the necessary height is minimized.

Separate lifting arrangements are usually required for overhaul of the turbocharger since the crane travel is limited by the exhaust pipe. A chain block on a rail located over the turbocharger axis is recommended.

18.2.3 Maintenance platforms

In order to enable efficient maintenance work on the engine, it is advised to build the maintenance platforms on recommended elevations. The width of the platforms should be at minimum 800 mm to allow adequate working space. The surface of maintenance platforms should be of non-slippery material (grating or chequer plate).

NOTE

Working Platforms should be designed and positioned to prevent personnel slipping, tripping or falling on or between the walkways and the engine

18.3 Transportation and storage of spare parts and tools

Transportation arrangement from engine room to storage and workshop has to be prepared for heavy engine components. This can be done with several chain blocks on rails or alternatively utilising pallet truck or trolley. If transportation must be carried out using several lifting equipment, coverage areas of adjacent cranes should be as close as possible to each other.

Engine room maintenance hatch has to be large enough to allow transportation of main components to/from engine room.

It is recommended to store heavy engine components on slightly elevated adaptable surface e.g. wooden pallets. All engine spare parts should be protected from corrosion and excessive vibration.

On single main engine installations it is important to store heavy engine parts close to the engine to make overhaul as quick as possible in an emergency situation.

18.4 Required deck area for service work

During engine overhaul some deck area is required for cleaning and storing dismantled components. Size of the service area is dependent of the overhauling strategy chosen, e.g. one cylinder at time, one bank at time or the whole engine at time. Service area should be plain steel deck dimensioned to carry the weight of engine parts.

18.4.1 Service space requirement for the in-line engine

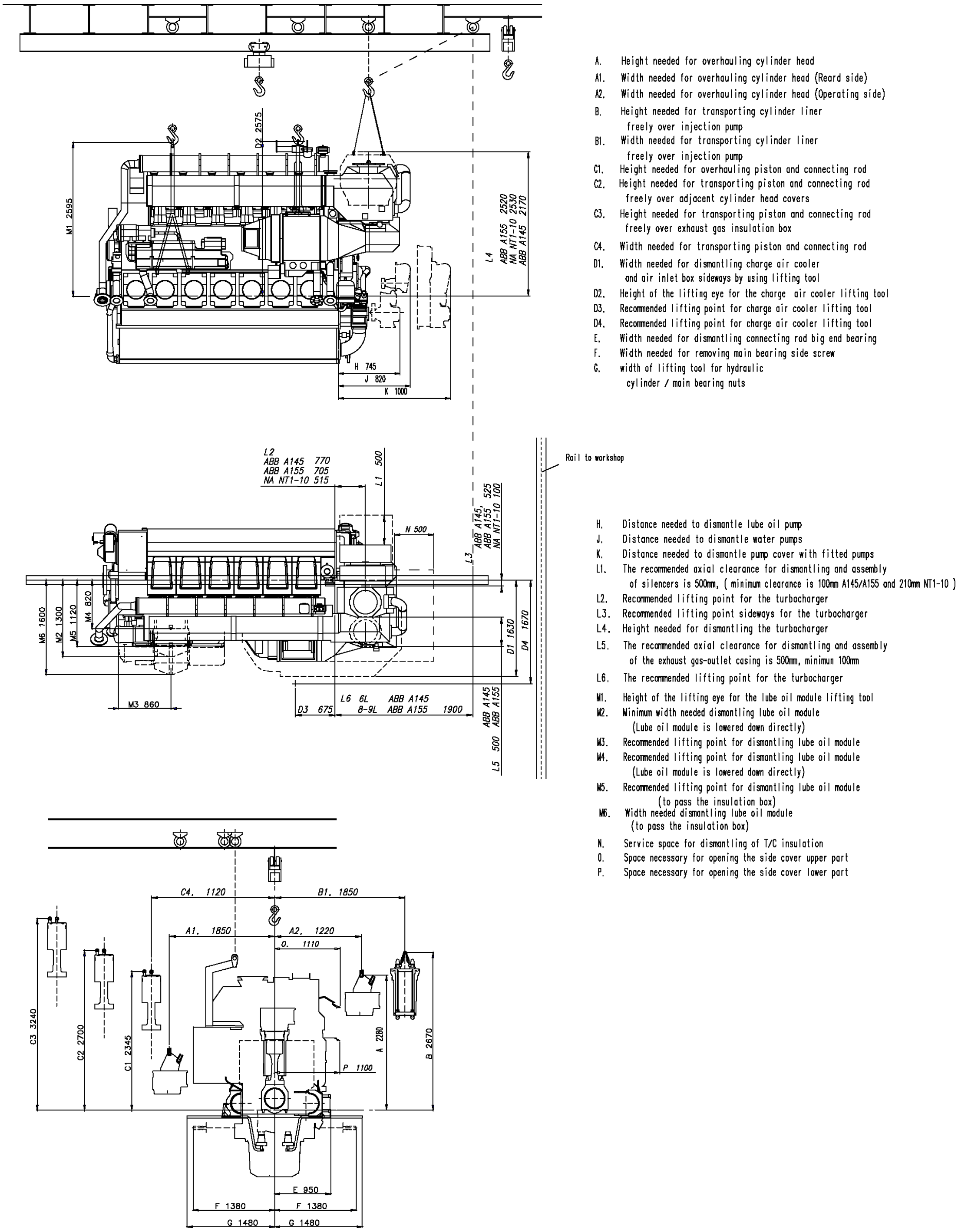


Fig 18-10 Service space requirement, turbocharger in free end (DAAF070676B)

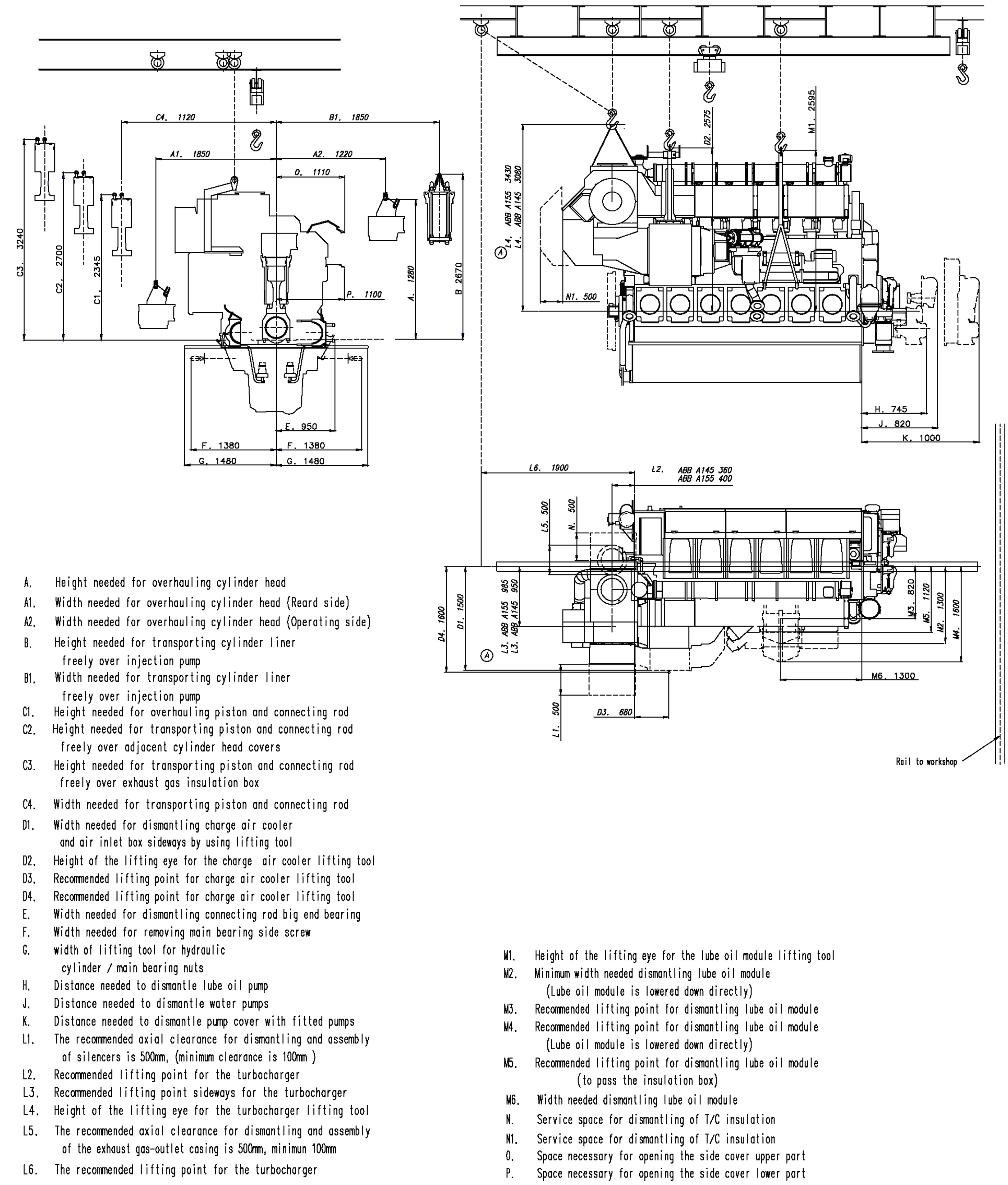


Fig 18-11 Service space requirement, turbocharger in driving end (DAAF088437B)

18.4.2 Service space requirement for the V-engine

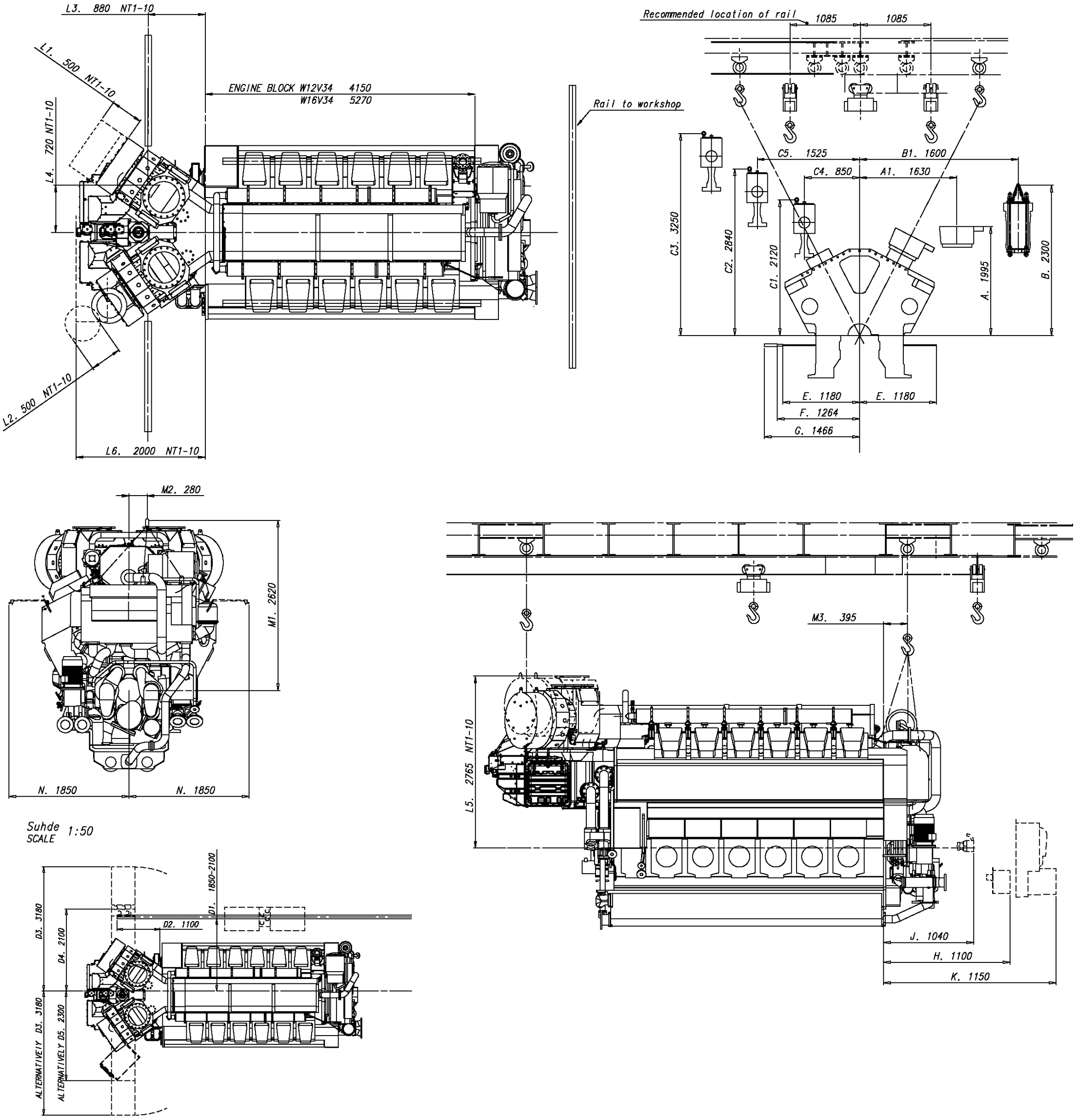


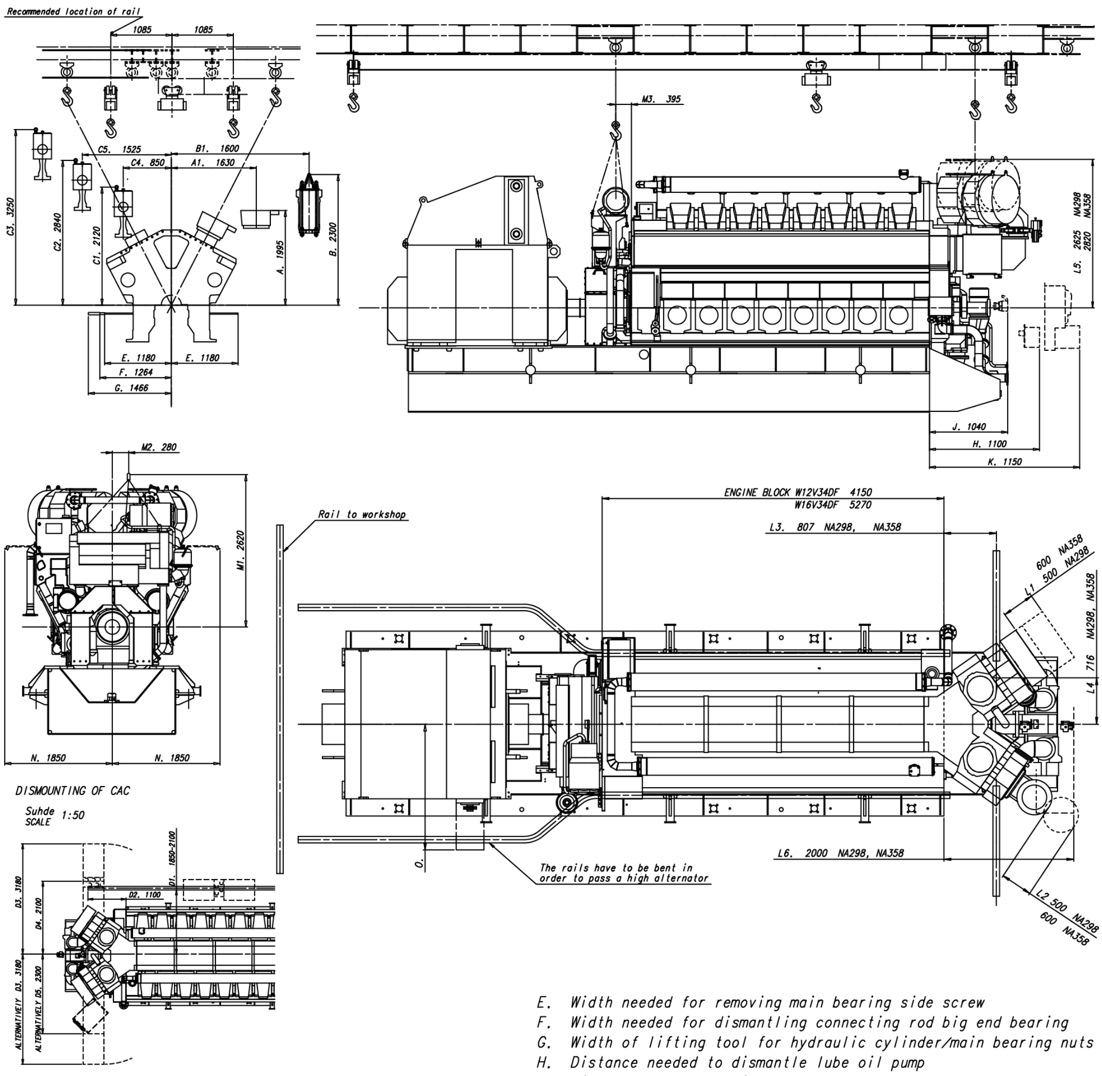
Fig 18-12 Service space requirement, turbocharger in driving end (DAAF308339)

Table 18-1 Positions in space requirement drawing (DAAF308339)

Pos	Description
A	Height needed for overhauling cylinder head
A1	Width needed for overhauling cylinder head
B	Heigth needed for overhauling cylinder liner
B1	Width needed for overhauling cylinder liner
C1	Height needed for overhauling piston and connecting rod
C2	Height needed for transporting piston and connecting rod freely over adjacent cylinder head covers
C3	Height needed for transporting piston and connecting rod freely over exhaust gas insulating box
C4, 5	Width needed for transporting piston and connecting rod
D1	Recommended location of rail for removing the CAC either on A- or B-bank
D2	Recommended location of starting point for rails
D3	Width needed for dismantling the whole CAC either from A-bank or B-bank

D4	Minimum width needed for dismantling CAC from B-bank when CAC is divided into 3 parts before turning 90º. (Pressure test in place)
D5	Minimum width needed for dismantling CAC from A-bank when CAC is divided into 3 parts before turning. (Pressure test in place)
E	Width needed for removing main bearing side screw
F	Width needed for dismantling connecting rod big end bearing
G	Width of lifting tool for hydraulic cylinder/main bearing nuts
H	Distance needed to dismantle lube oil pump
J	Distance needed to dismantle water pump
K	Distance needed to dismantle pump cover with fitted pumps
L1	The recommended axial clearance for dismantling and assembling of silencer is 500mm, minimum clearance is 120mm for NT1-10 The given dimension for L1 includes the minimum maintenance space
L2	The recommended axial clearance for dismantling and assembling of suction branches is 500mm, minimum clearance is 120mm for NT1-10 The given dimension for L2 includes the minimum maintenance space
L3	Recommended lifting point for the turbocharger
L4	Recommended lifting point sideways for the turbocharger
L5	Height needed for dismantling the turbocharger
L6	Recommended space needed to dismantle insulation, (CAC overhaul)
M1	Height of lube oil module lifting tool eye
M2	Width of lube oil module lifting tool eye
M3	Width of lube oil module lifting tool eye
N	Space necessary for opening the side cover

18.4.3 Service space requirement for the genset



A. Hight needed for overhauling cylinder head
A1. Width needed for overhauling cylinder head
B. Hight needed for overhauling cylinder liner
B1. Width needed for overhauling cylinder liner
C1. Height needed for overhauling piston and connecting rod
C2. Height needed for transporting piston and connecting rod freely over adjacent cylinder head covers
C3. Height needed for transporting piston and connecting rod freely over exaust gas insulation box
C4. Width needed for transporting piston and connecting rod
C5. Width needed for transporting piston and connecting rod freely over adjacent cylinder head covers
D1. Recommended location of rail for removing the CAC either on A- or B-bank.
D2. Recommended location of starting point for rails.
D3. Width needed for dismantling the whole CAC either from A-bank or B-bank. (Advantage: CAC can be pressure tested before assembly)
D4. Minimum width needed for dismantling CAC from B-bank when CAC is divided into 3 parts before turning 90°. (Pressure test in place)
D5. Minimum width needed for dismantling CAC from A-bank when CAC is divided into 3 parts before turning. (Pressure test in place)
E. Width needed for removing main bearing side screw
F. Width needed for dismantling connecting rod big end bearing
G. Width of lifting tool for hydraulic cylinder/main bearing nuts
H. Distance needed to dismantle lube oil pump
J. Distance needed to dismantle water pumps
K. Distance needed to dismantle pump cover with fitted pumps
L1. The recommended axial clearance for dismantling and assembling of silencer is 500mm for NA298, 600mm for NA358. Minimum clearance is 120mm for NA298, 150mm for NA358 The given dimension for L1 includes the minimum maintenance space.
L2. The recommended axial clearance for dismantling and assembling of suction branches is 500mm for NA298, 600m for NA358 minimum clearance is 120mm for NA298, NA358 The given dimension for L2 includes the minimum maintenance space .
L3. Recommended lifting point for the turbocharger
L4. Recommended lifting point sideways for the turbocharger
L5. Height needed for dismantling the turbocharger
L6. Recommended space needed to dismantle insulation, (CAC overhaul)
M1. Height of lube oil module lifting tool eye
M2. Width of lube oil module lifting tool eye
M3. Width of lube oil module lifting tool eye
N. Space necessary for opening the side cover
O. Service space for generator cooler, depending on generator type.

A

Fig 18-13 Service space requirement, genset (DAAF066517A)

19. Transport Dimensions and Weights

19.1 Lifting of main engines

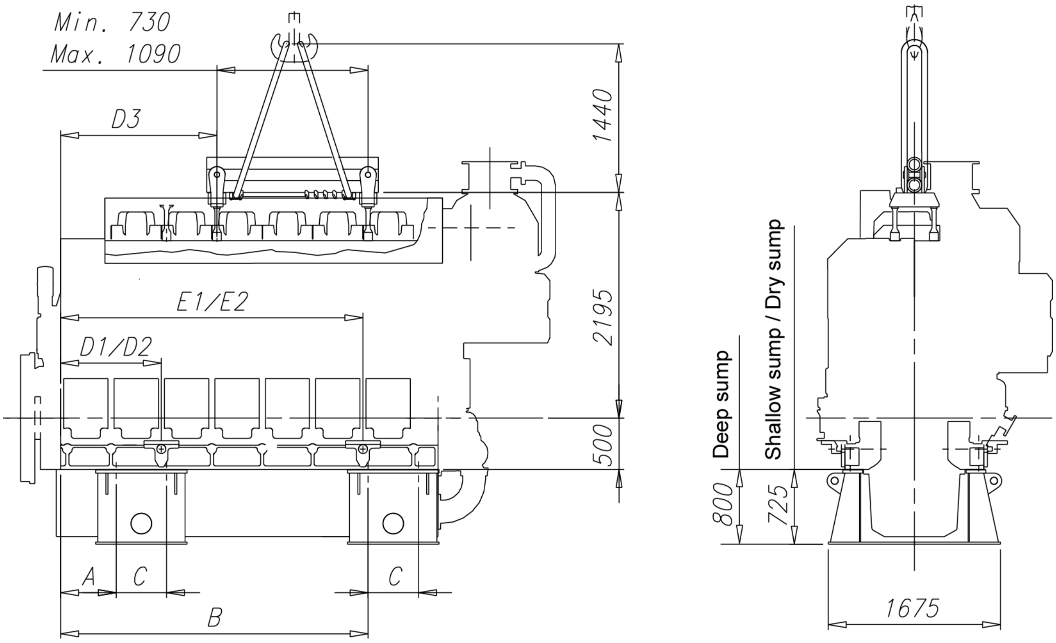


Fig 19-1 Lifting of main engines, in-line engines (DAAF068506A)

Engine	A	B	C	D1*	D2*	D3	E1*	E2*
W 6L34DF	540	2990	490	980	980	1520	2940	2940
W 8L34DF	540	3970	490	490	980	2010	3430	3920
W 9L34DF	540	4460	490	490	980	2010	3920	4410

All dimensions in mm. Transport bracket weight: 890 kg.

* 1 = Rear side (B-bank)

2 = Operating side (A-bank)

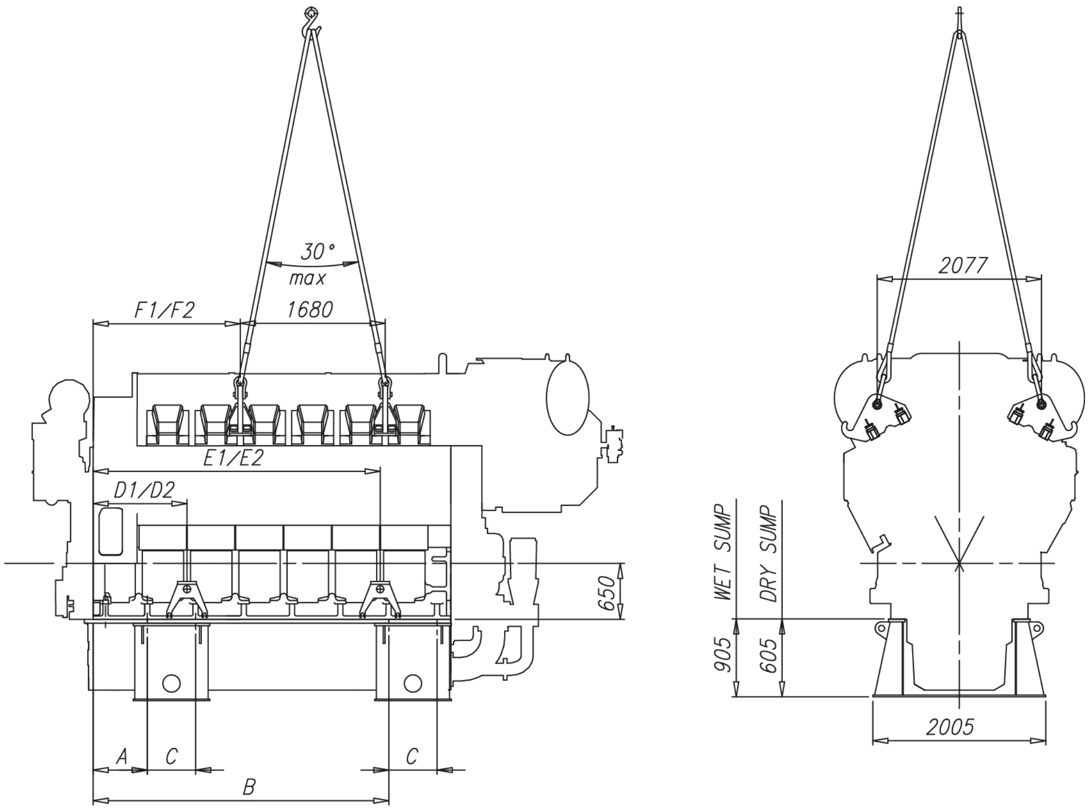


Fig 19-2 Lifting of main engines, V-engines (DAAF068506)

Engine	A	B	C	D1*	D2*	E1*	E2*	F1*	F2*
W 12V34DF	630	3430	560	1090	530	3330	3330	1706	1594
W 16V34DF	630	4550	560	1090	530	4450	4450	2266	2154

All dimensions in mm. Transport bracket weight: dry oil sump = 935 kg, wet oil sump = 1060 kg.

* 1 = Rear side (B-bank)

2 = Operating side (A-bank)

19.2 Lifting of generating sets

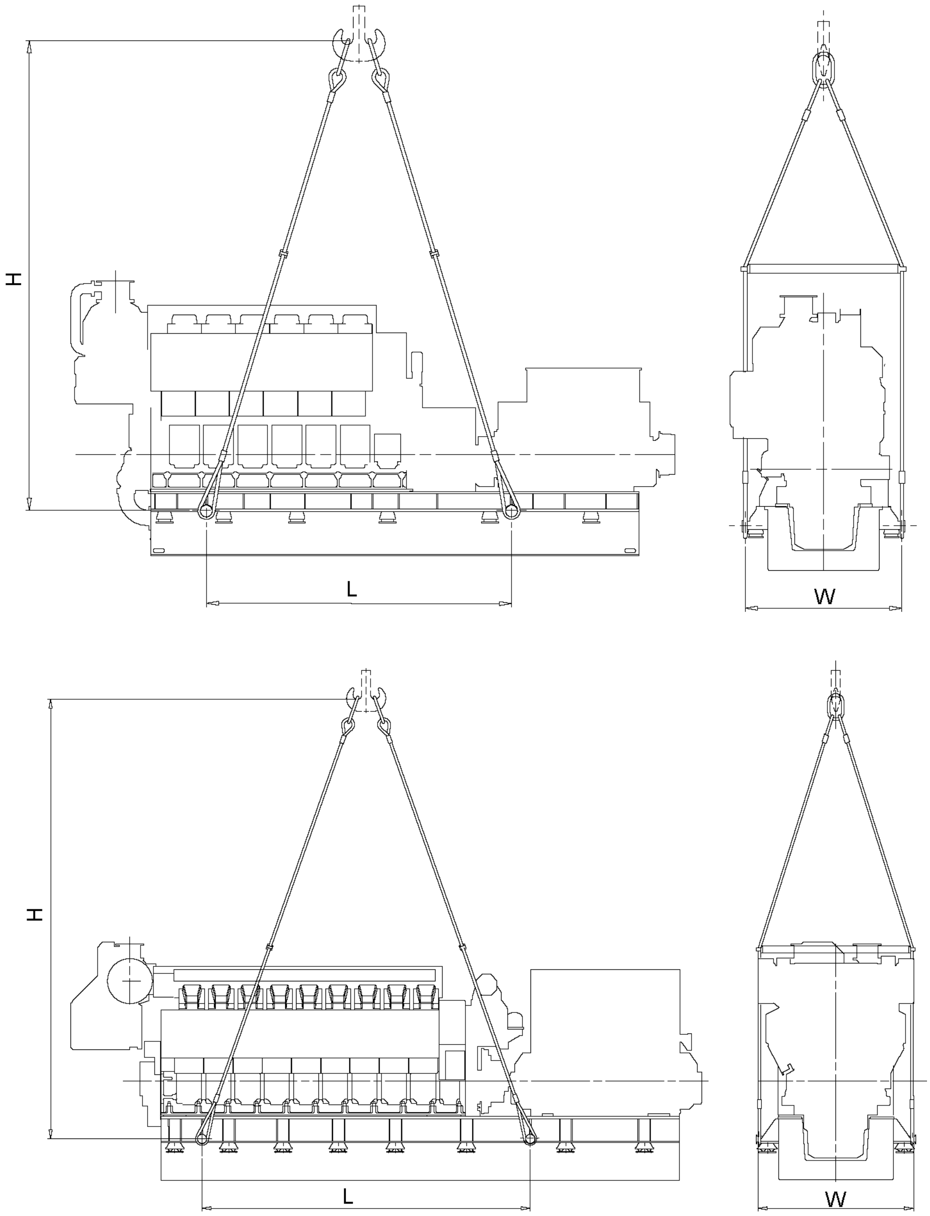
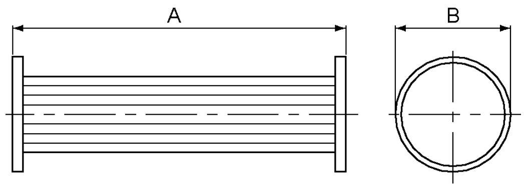


Fig 19-3 Lifting of generating sets (DAAE083966A, -69B)

Engine	H [mm]	L [mm]	W [mm]
W L34DF	6595...6685	4380...6000	2240...2645
W V34DF	6900...9400	5500...9400	2940...3275

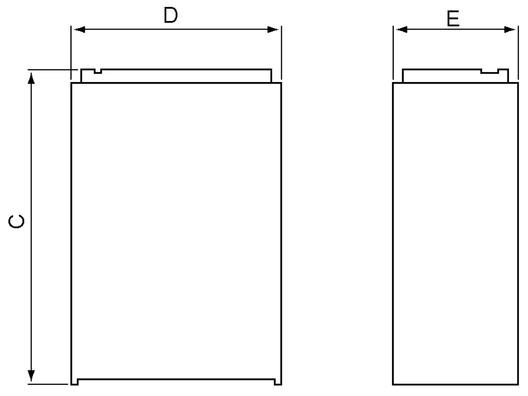
19.3 Engine components

Table 19-1 Lubricating oil insert (DAAE083974)



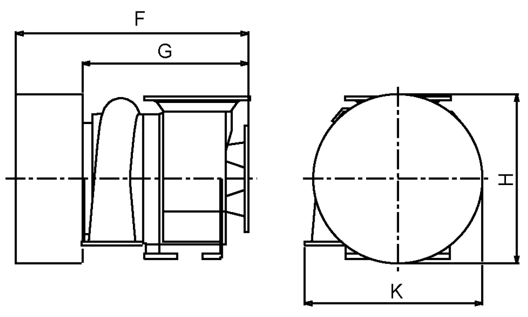
Engine	Dimensions [mm]		Weight [kg]
	A	B	
W 6L34DF	650	369.4	105
W 9L34DF	1140	369.4	120
W 12V34DF	1338	479.4	250
W 16V34DF	1338	479.4	250

Table 19-2 Charge air cooler insert (DAAE083974)



Engine	Dimensions [mm]			Weight [kg]
	C	D	E	
W 6L34DF	963	672	436	500
W 9L34DF	963	790	594	500
W 12V34DF	2056	600	600	850
W 16V34DF	2056	600	600	950

Table 19-3 Turbocharger (DAAE083974)



Engine	Dimensions [mm]				Weight (kg)
	F	G	H	K	
W 6L34DF	1017	663	861	766	773
W 9L34DF	1017	663	861	766	783
W 12V34DF	1017	663	861	766	2 x 773
W 16V34DF	1017	663	861	766	2 x 783

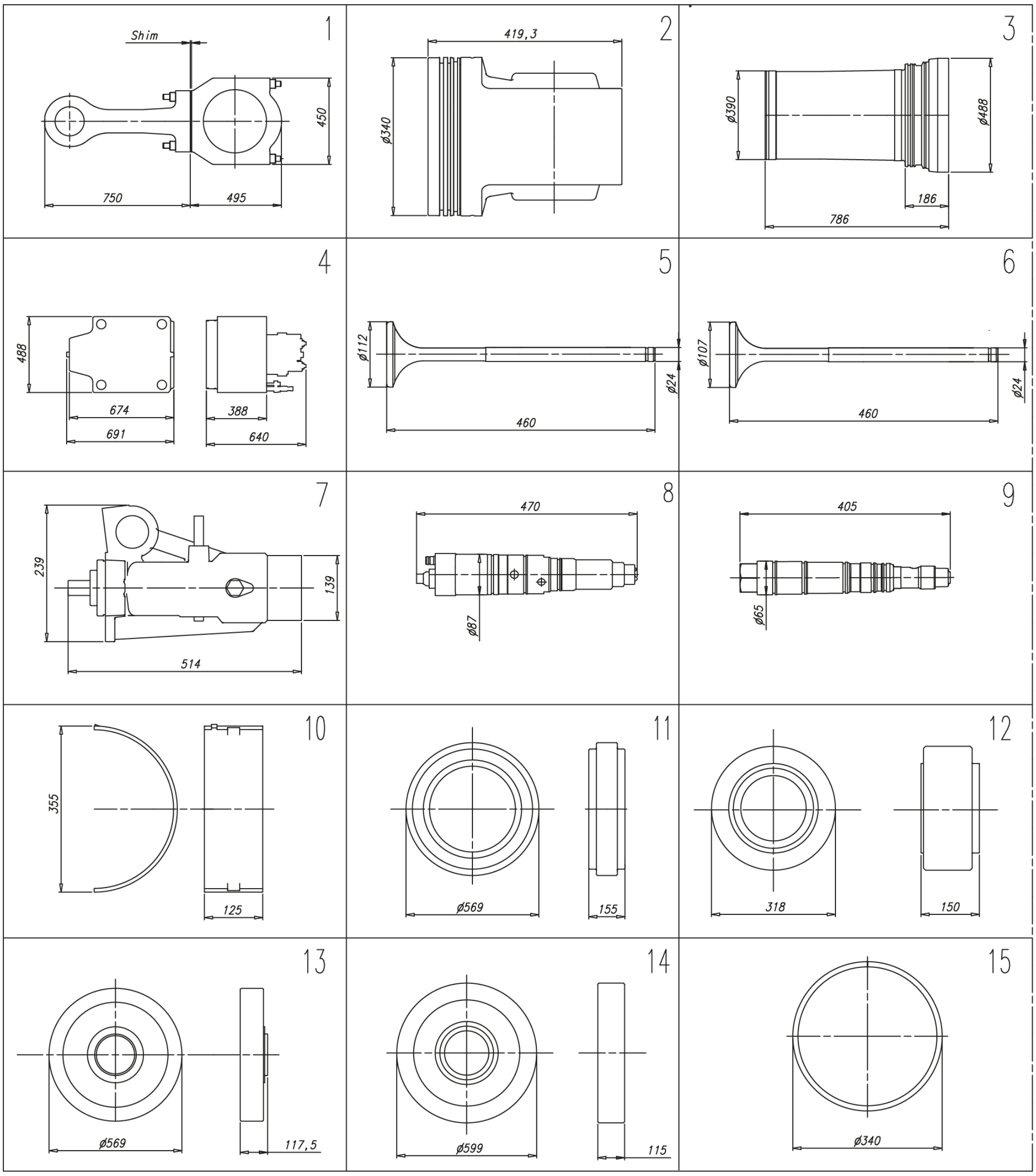


Fig 19-4 Major spare parts (DAAF073204)

Item no	Description	Weight [kg]	Item No	Description	Weight [kg]
1	Connecting rod	157	9	Starting valve	6.1
2	Piston	107	10	Main bearing shell	7.5
3	Cylinder liner	223	11	Split gear wheel	121
4	Cylinder head	376	12	Small intermediate gear	49
5	Inlet valve	3	13	Large intermediate gear	107
6	Exhaust valve	2.9	14	Camshaft gear wheel	132
7	Injection pump	50	15	Piston ring set	1.5
8	Injection valve	15.5		Piston ring	0.5

21. ANNEX

21.1 Unit conversion tables

The tables below will help you to convert units used in this product guide to other units. Where the conversion factor is not accurate a suitable number of decimals have been used.

Length conversion factors

Convert from	To	Multiply by
mm	in	0.0394
mm	ft	0.00328

Mass conversion factors

Convert from	To	Multiply by
kg	lb	2.205
kg	oz	35.274

Pressure conversion factors

Convert from	To	Multiply by
kPa	psi (lbf/in^2)	0.145
kPa	lbf/ft^2	20.885
kPa	inch H_2O	4.015
kPa	foot H_2O	0.335
kPa	mm H_2O	101.972
kPa	bar	0.01

Volume conversion factors

Convert from	To	Multiply by
m^3	in^3	61023.744
m^3	ft^3	35.315
m^3	Imperial gallon	219.969
m^3	US gallon	264.172
m^3	l (litre)	1000

Power conversion

Convert from	To	Multiply by
kW	hp (metric)	1.360
kW	US hp	1.341

Moment of inertia and torque conversion factors

Convert from	To	Multiply by
kgm^2	lbft^2	23.730
kNm	lbf ft	737.562

Fuel consumption conversion factors

Convert from	To	Multiply by
g/kWh	g/hph	0.736
g/kWh	lb/hph	0.00162

Flow conversion factors

Convert from	To	Multiply by
m^3/h (liquid)	US gallon/min	4.403
m^3/h (gas)	ft^3/min	0.586

Temperature conversion factors

Convert from	To	Multiply by
°C	F	F = 9/5 *C + 32
°C	K	K = C + 273.15

Density conversion factors

Convert from	To	Multiply by
kg/m^3	lb/US gallon	0.00834
kg/m^3	lb/Imperial gallon	0.01002
kg/m^3	lb/ft^3	0.0624

21.1.1 Prefix

Table 21-1 The most common prefix multipliers

Name	Symbol	Factor	Name	Symbol	Factor	Name	Symbol	Factor
tera	T	10^{12}	kilo	k	10^3	nano	n	10^{-9}
giga	G	10^9	milli	m	10^{-3}			
mega	M	10^6	micro	μ	10^{-6}			

21.2 Collection of drawing symbols used in drawings

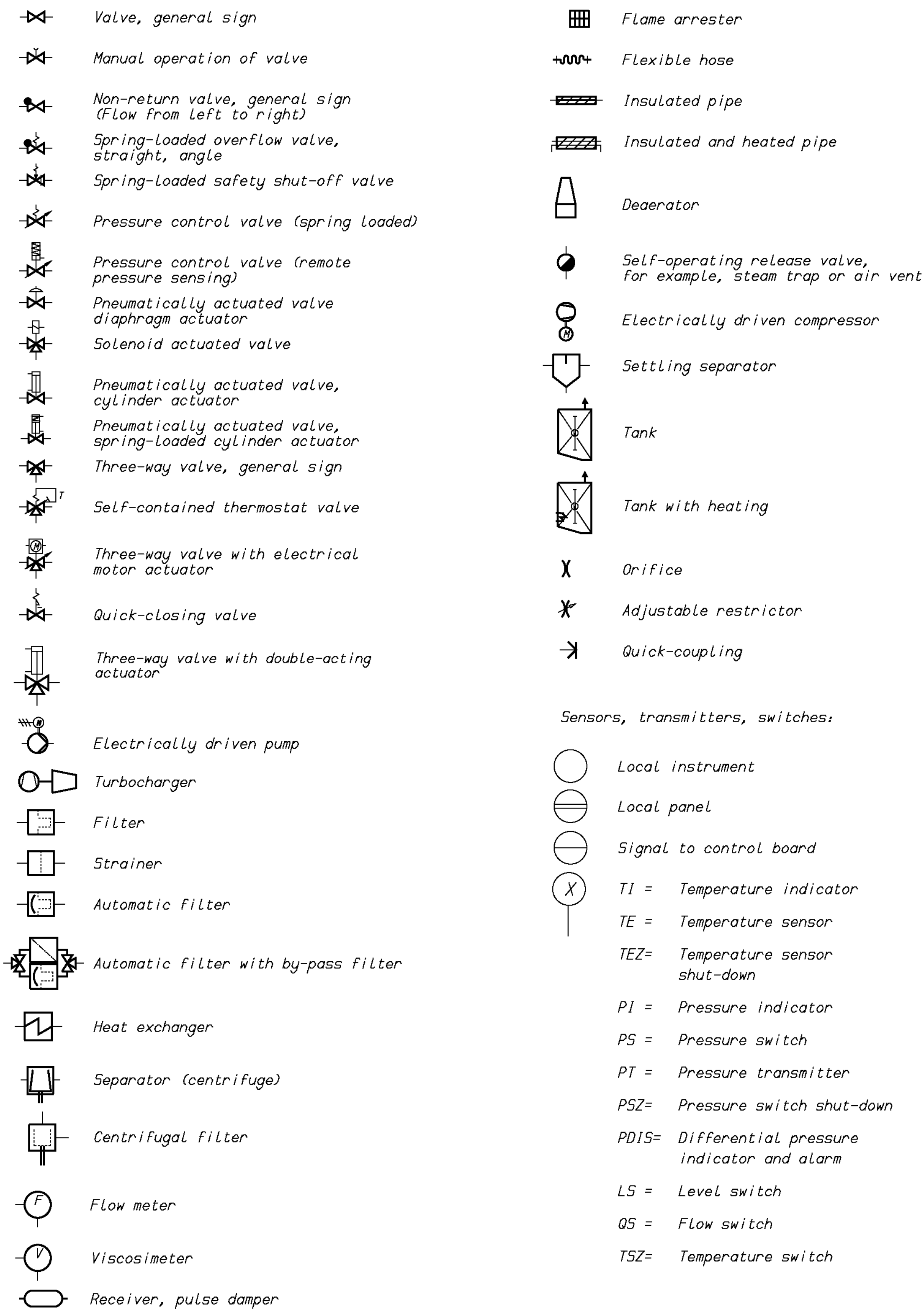


Fig 21-1 List of symbols (DAAE000806c)

10.3. ANEXO III DOCUMENTACIÓN TIMONES DE RENDIMIENTO

Promas

Promas offers increased propulsive efficiency and improved manoeuvrability by adapting the propeller and rudder into one propulsive unit. It is suitable for conventional single and twin screw ships.

Each installation comprises a twisted full spade rudder with a Costa bulb that is smoothly connected to the propeller by a hubcap, and a propeller design adapted to the rudder.

A well designed twist adapts the rudder to the rotation of the propeller slipstream and reduces the angle of attack on the rudder's leading edge. This gives a more efficient rudder profile with lower drag and better recovery of rotational energy from the propeller slipstream.

Promas

Promas + nozzle

Promas

Promas integrates the propeller, a hubcap, rudder bulb and the rudder itself into a single hydrodynamic efficient unit.

A tapered hubcap fitted to the propeller hub leads the waterflow onto a bulb which forms part of the spade rudder. The rudder has a twisted leading edge, optimized for the flow from the propeller, which converts to into additional forward thrust some of the swirl energy in the slipstreams that is normally lost.

The result is an increase in propulsive efficiency of up to 8 per cent depending on the application, leading to reduced fuel consumption and emissions. Large steering forces can also be developed.

Promas has been developed using the latest CFD technologies. As the risk of hub vortex cavitation is removed., the radial distribution of hydrodynamic loads on the propeller blades can be modified, reducing tip loading and helping to limit the intensity of blade pressure pulses (up to 25 per cent) and associated noise and vibration.

Promas + nozzle

Developed specifically for offshore vessels. A new nozzle, propeller, hubcap, bulb and rudder profile combine to maximise free-running efficiency and improve bollard pull, typically by 5 – 8 per cent. Water flow leaving the nozzle passes over the special profile rudder to provide high steering forces yet minimum drag.

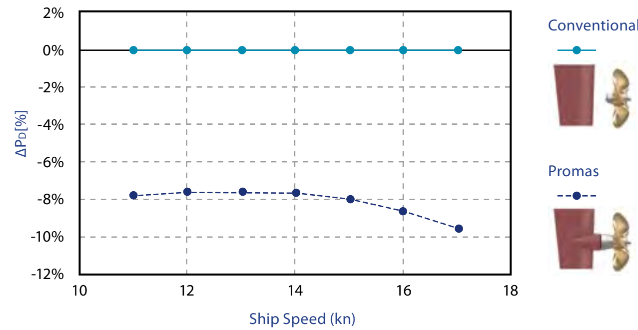
Key features:

- Propeller and rudder designed as a single system for optimum efficiency
- Propulsive efficiency increased by 3 – 8 per cent
- Improved low speed manoeuvrability
- Improved possibility for low pressure pulse/low noise propeller design
- Almost as easy to install as a conventional propeller-rudder system
- Nozzle option can reduce fuel consumption in transit by 15 – 20 per cent
- Simple and robust with short payback time

Propulsive efficiency improved

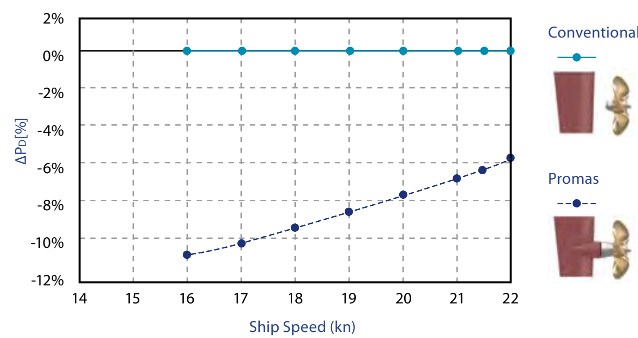
In general, the Promas efficiency gain is in the region of 3 – 8 per cent for single screw, and 2 – 6 per cent for twin screw vessels. A good rudder design with optimised profile shape, positioning of the rudder in the slipstream and skeg design can increase the propulsive efficiency by an additional 2 per cent. Comparison tests between a conventional propeller-rudder system and Promas are shown in the graphs below.

Relative power delivered vs. Ship speed - Single screw vessel



The rudder area, profile shape and position are identical for the conventional and Promas cases in the graphs below. So the increase in efficiency shown is the pure effect of the bulb, hubcap, rudder twist and adapted propeller design.

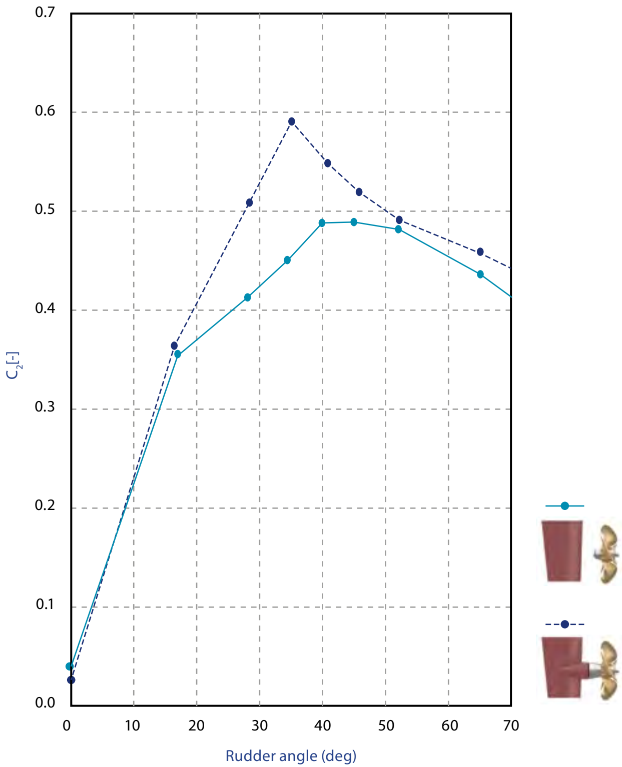
Relative power delivered vs. Ship speed - Twin screw vessel



Improved manoeuvring at low speed

At low speed manoeuvring ie. harbour manoeuvring, a maximum side force and a maximum rudder drag is important. The graph below shows the non-dimensional lift against rudder angle for a single screw vessel.

Rudder lift vs. Rudder angle - bollard pull (equivalent to low speed manoeuvring)



Promas Lite

Promas Lite is a version of the successful Promas system that can be easily fitted to vessels already in service. The installation is simple with only three areas of modification:

- Welding a prefabricated bulb in position on the existing rudder
- Bolting the hubcap to the propeller hub
- Fitting of a new propeller or reblading the original one

Improving propulsive efficiency is key to reducing fuel burn and emissions. Promas Lite installations on vessels operating significantly off their original design speed should provide an efficiency improvement in the region of 5 – 15 per cent. Recent installations on twin screw cruise vessels have demonstrated efficiency improvements within these guidelines giving a payback period of well under two years. The improvement it delivers in propulsive efficiency means that engine loads are reduced, which also helps to lower wear and tear on the engine.

Key features:

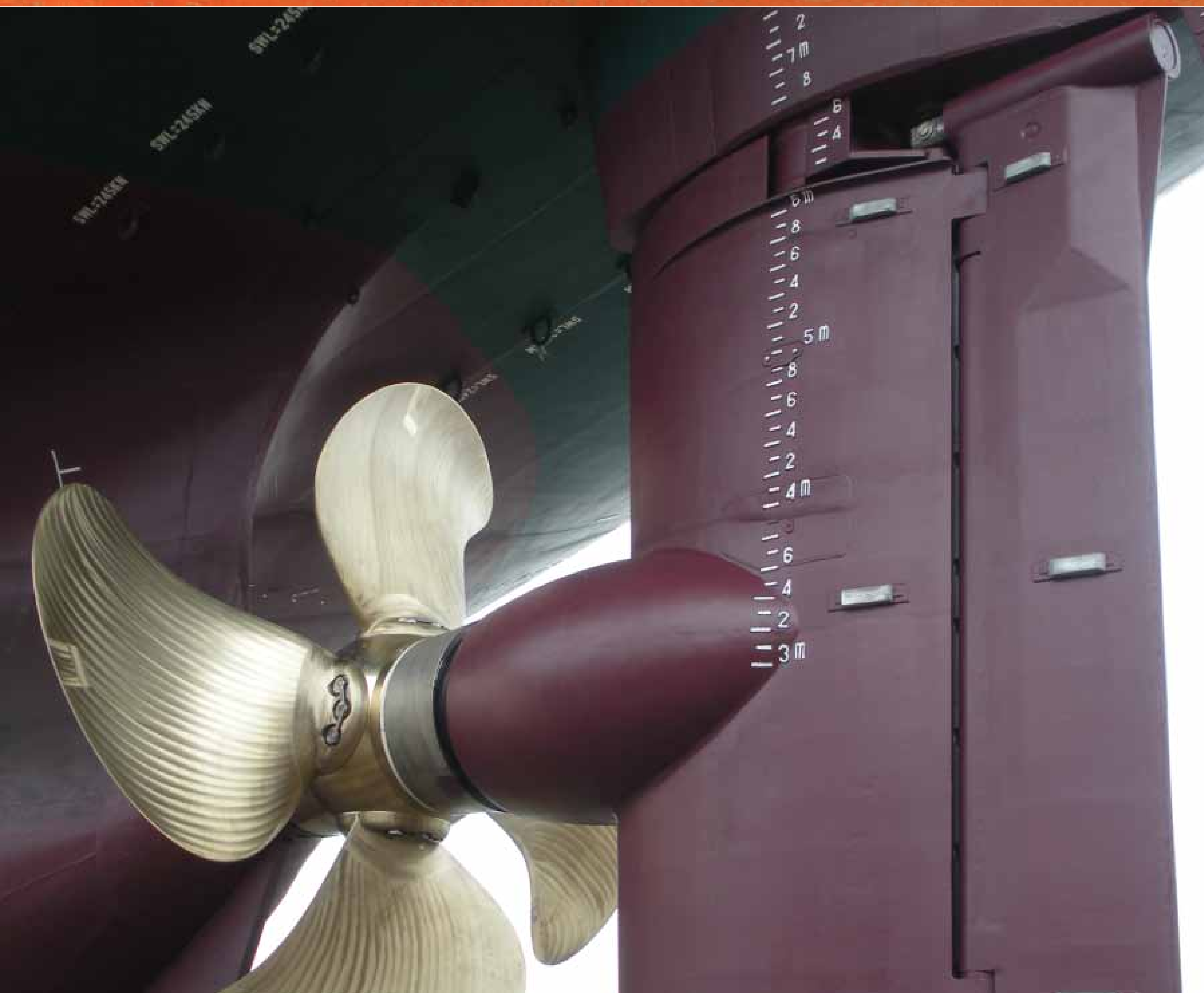
- Reduced fuel consumption of between 5 – 15 per cent
- Lower exhaust emissions
- Short payback time
- Simple and quick installation (7 – 10 days)

Before installation.

After installation.

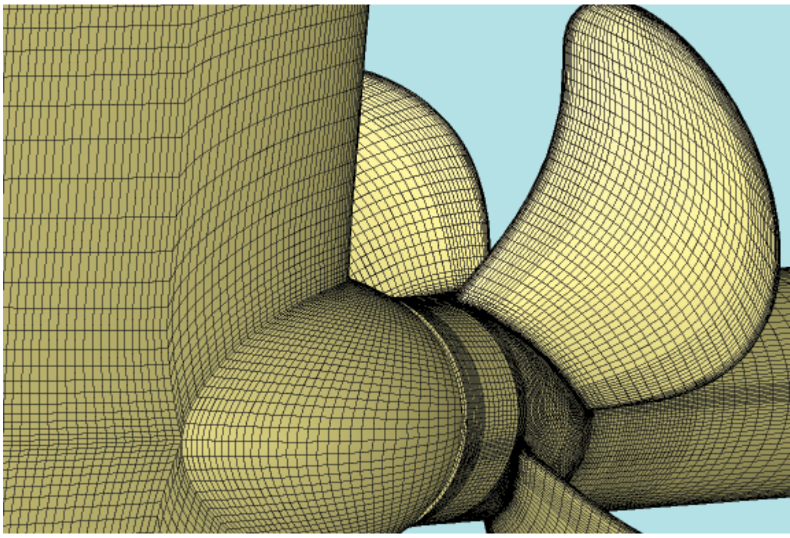
WÄRTSILÄ

WÄRTSILÄ Rudders



Courtesy of Finnlines Oy, photographed by Hannu Laakso.

Energopac incorporated in model tests, photographed by HSVA.



CFD capabilities within Wärtsilä are state-of-the-art.

OPTIMIZING ENERGY EFFICIENCY

Wärtsilä is continuously looking to improve the energy efficiency of its propulsion solutions. In so doing, we aim to reduce fuel consumption, lower the operational costs of seagoing vessels, and, of course, cut back on emissions. We design our solutions to meet specific customer requirements, utilizing our extensive marine industry experience and state-of-the-art, modern techniques, such as CFD (Computational Fluid Dynamics).

Our high efficiency rudder technology dates back to the 1990s when Wärtsilä started making energy-saving rudders. Since then more than 30 vessels have been equipped with our energy-saving rudders, and all of them have proven to be very successful in saving fuel. For example, the fuel consumption of a series of chemical tankers has been reduced by 5% sailing at 17 knots. Another example is a series of general cargo vessels, with fuel savings of 4% at 23 knots.

In 2009, two Energopac systems were delivered to a newbuilding project, and in early 2010 the first Energopac retrofit was successfully installed. Another 12 Energopac systems are currently on order for delivery in 2010 and 2011.

The Wärtsilä Propeller-Rudder System, Energopac has been designed to include the following benefits:

- improved energy efficiency and reduced fuel consumption, thanks to integrated propeller and rudder design
- excellent manoeuvrability
- lower vibration levels and higher comfort onboard
- reduced levels of emissions

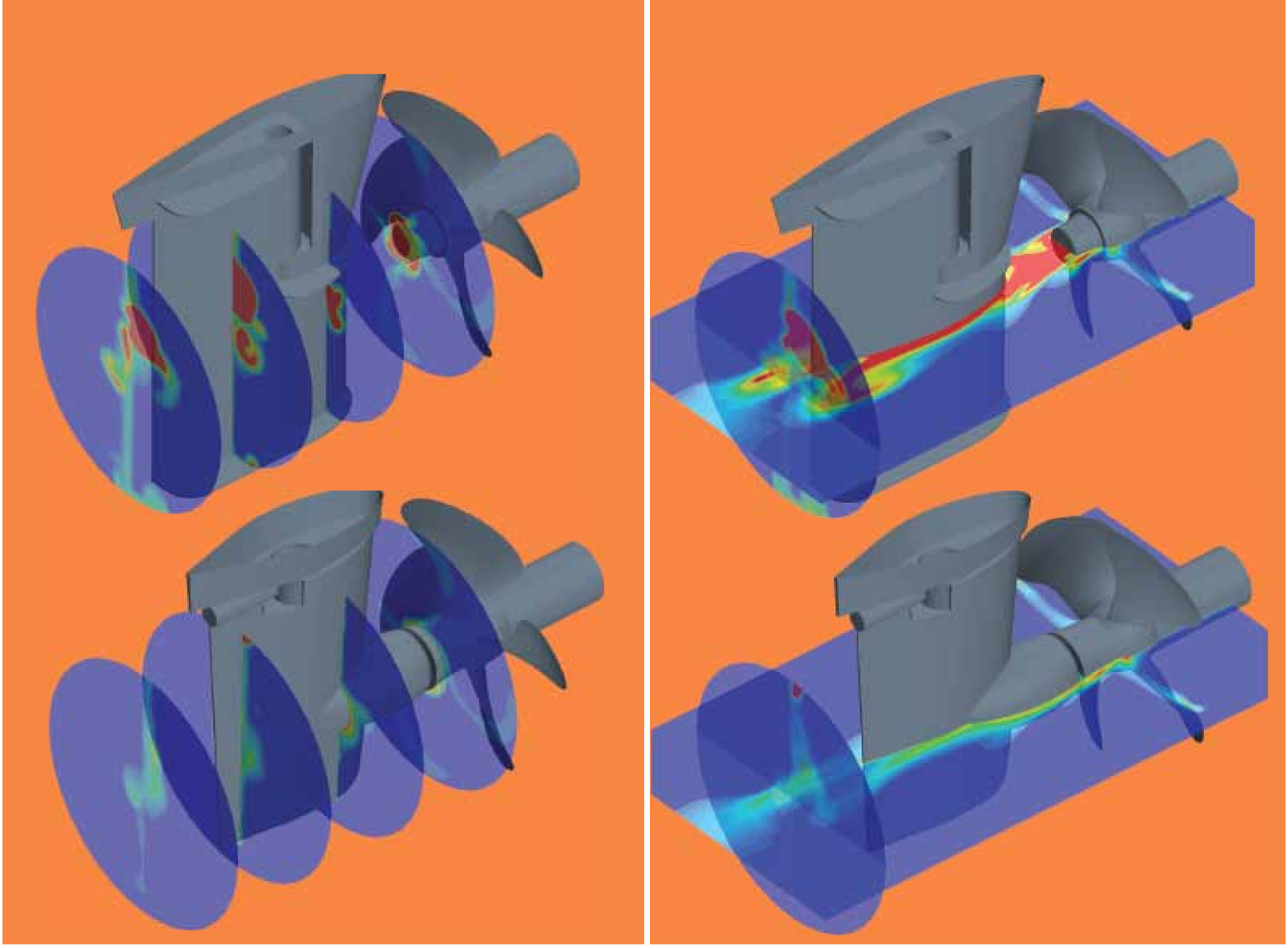
ENERGOPAC INTEGRATED PROPULSION AND MANOEUVRING PERFORMANCE

A vessel's power efficiency level is dependent upon interaction between all the main components. To achieve optimal performance, these need to form a single integrated design. This also holds true for the interaction between the vessel's propeller and rudder.

Energopac is an optimized propulsion and manoeuvring solution for coastal and seagoing vessels. Its key objective is to reduce a vessel's fuel consumption and CO_2 emissions through integrating the propeller and rudder design. Energopac is tailored for each and every vessel to meet the customer's specific requirements, and can thus be optimized for energy efficiency, without compromising manoeuvrability or comfort.

DESIGNED FOR PERFECTION

Energopac was developed through co-operation between Wärtsilä's propulsion specialists, and the rudder experts at Becker Marine Systems. The combined experience and track record of these companies in optimizing ship efficiency is extensive, and is widely appreciated throughout the industry.



CFD images show that streamlining the flow significantly reduces separation losses from the propeller hub.

The combination of propulsion expertise and vessel manoeuvring know-how forms the backbone of this co-operation. The result is a sophisticated propulsion and manoeuvring solution, designed and optimized for energy efficiency.

The amount of fuel savings, when applying Energopac, depends very much on the vessel type, its operational profile, and on the reference propeller and rudder. We can give a good estimation of the potential annual fuel savings after having examined the vessel's design, together with alternative propulsion solutions, in detail.

Energopac is then optimized according to the specifics of the vessel. In-house, state-of-the-art CFD (Computational Fluid Dynamics) capabilities are used for the engineering. CFD makes it possible to customize the equipment for any vessel, and to tailor it to match the vessel's operational profile and other specific needs.

ENERGOPAC IN DETAIL

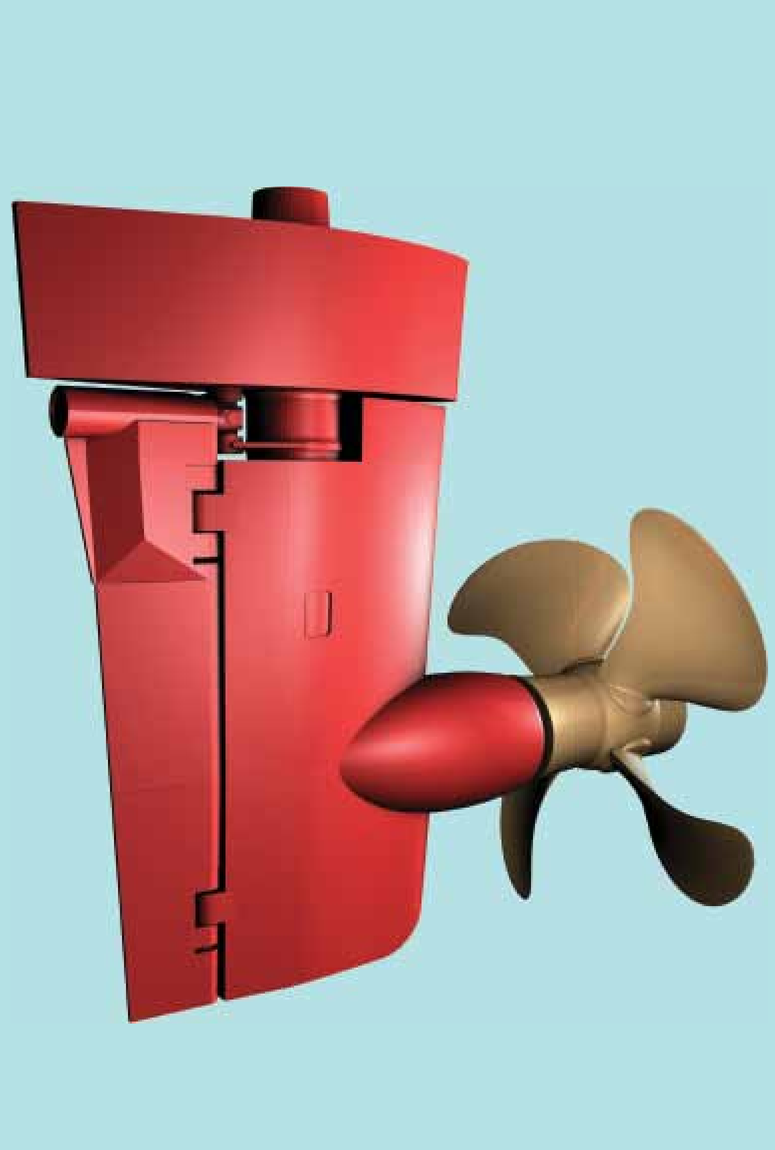
Energopac comprises several components, which vary in design and dimension, and which are modified to integrate with each other to serve the vessel in the most effective way possible. These include the propeller, a streamlined fairing cap, and a rudder system with a rudder bulb for efficiency. The interacting parts of the system are co-designed with Becker Marine Systems

Energopac includes a sophisticated full-spade flap rudder. Firstly, this provides excellent rudder balance and manoeuvring performance; and secondly it allows for a smaller overall rudder blade area, which results in lower rudder drag. Furthermore, it requires only relatively small steering angles to keep the vessel on course when in transit ensuring that the rudder bulb stays in the shadow of the fairing cap wake in steering condition.

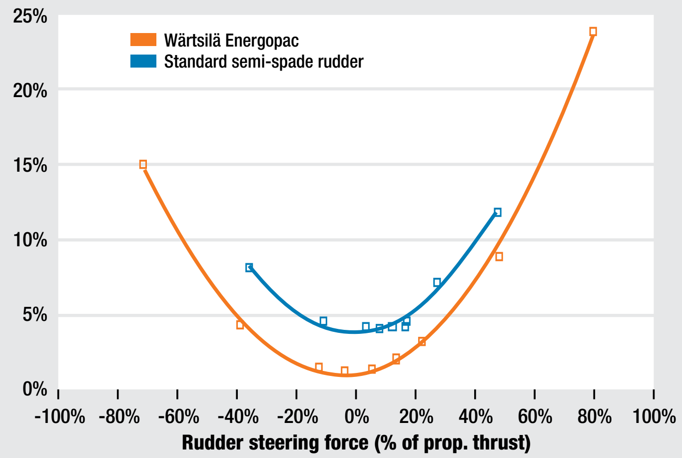
The KSR (King Support Rudder) system patented by Becker is used to construct a slim rudder profile, thereby minimizing resistance and providing a strong and stiff rudder support. The KSR support system enables the building of full spade rudders of any size, and to suit the highest ship speeds. This allows Energopac to be installed on any size vessel, even large or higher-speed vessels, which is truly unique. The asymmetric profile of the twisted leading edge rudder is aligned perfectly with the propeller's rotating slipstream. This results in less acceleration at the rudder leading edge, making the rudder less prone to propeller induced cavitation and recovering a part of the rotational energy of the slipstream, improving the propulsion efficiency of the propeller-rudder system.

The fairing cap and rudder bulb are specifically designed to reduce the separation losses behind the propeller hub, and to increase overall efficiency.

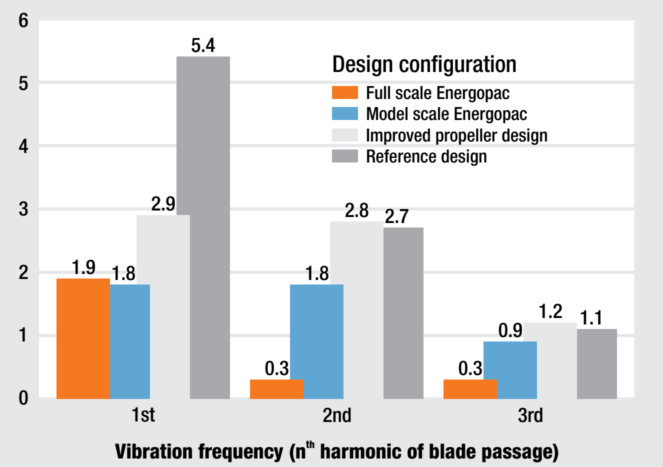
The propeller design is adapted to individual propulsion requirements. However, the design



Example arrangement of a typical Energopac solution, showing all the main components.



Energopac reduces flow separation behind the propeller hub and creates less drag than conventional rudder systems.



A standard propeller-rudder design compared with the improved design using Energopac. The reduction in pressure pulses, due to a more homogeneous water inflow into the propeller, results in lower vibration and increased comfort onboard the vessel.

is also optimized for both the rudder and the rudder bulb, thereby enabling even greater propeller efficiency when applying Energopac.

WORKING PRINCIPLES

By reducing the flow separation behind the propeller hub, Energopac effectively reduces the vessel's fuel consumption. Extended studies show that for the same course-keeping capabilities, Energopac creates less drag than conventional rudder systems.

The efficient design of the rudder bulb, particularly when integrated with the propeller and the rudder profile, streamlines the flow and significantly reduces separation losses from the propeller hub. The impact is most noticeable when a large part of the rudder bulb is behind the propeller hub. Thus, the highest fuel savings are achieved when the vessel is in transit, since the rudder angles required for maintaining course are minimal.

In course-keeping mode, Energopac has proven to save fuel more effectively than a conventional system. In particular, when using small forces for correction and steering to keep the vessel on course, the difference in rudder resistance is significant. The high-lift performance of Energopac requires smaller steering angles, and consequently reduces rudder resistance.

EFFIENCY AND COMFORT COMBINED

High performance propeller designs are often a compromise between increased efficiency and reduced vibration levels. The application of Energopac gives the vessel's designer or propulsion engineer greater freedom in optimizing opposing requirements. Energopac allows for a propeller design that meets both requirements! It will increase propulsion efficiency and/or reduce vibration levels, according to preference. At Wärtsilä we have the experience and know-how to advise our customers on the best solution for their specific combination of requirements.

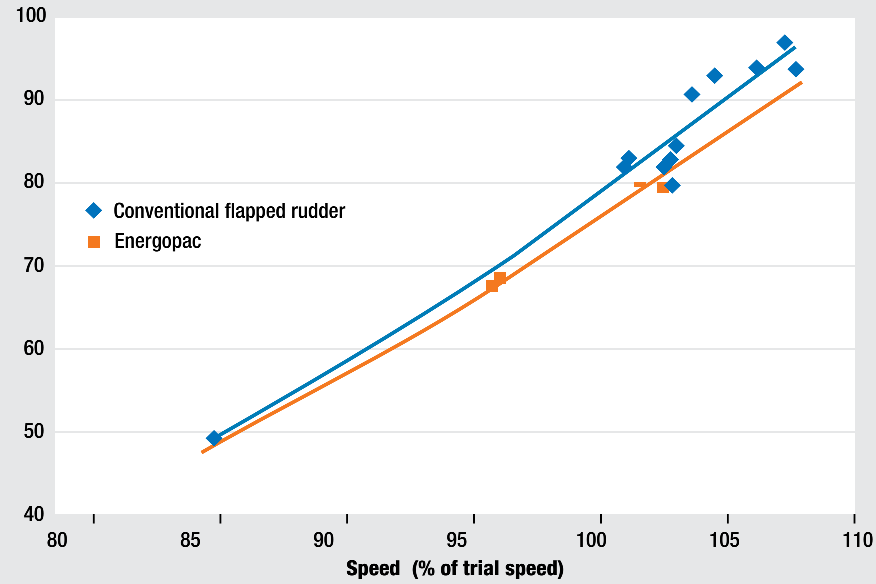
TYPICAL APPLICATIONS

Energopac will effectively reduce the operational costs for any vessel with a considerable share of free sailing time in its operational profile. It works very well for propellers with a relatively large propeller hub. The potential savings are largest for vessels having highly loaded controllable pitch propeller systems, such as RoRo-vessels, ferries, container / multipurpose vessels, and vessels with an ice class notation. On such ships with larger propeller hub diameters, the hub losses are significant with a conventional system. By applying Energopac, however, these losses can largely be avoided. Furthermore, Energopac will also show a favourable return on investment for any coastal or ocean-going vessels with considerable transit times.

SAVING WITH ENERGOPAC

Fuel savings with Energopac can be estimated for any application and are based on the vessel type, the operational profile of the vessel, and the reference propeller and rudder. Proven savings in the required power for a vessel's trial speed vary between 2% and 9%. As the actual reduction in power depends on the propulsion alternatives, the following examples give the annual savings in fuel costs for each percentage point in power reduction (based on fuel prices applicable in May 2010):

Case 1:
A 20,000 dwt cargo vessel with an 8000 kW main engine and a CPP installation will benefit greatly from Energopac in terms of power reduction. Depending on the operational profile of the vessel, the annual savings can be up to USD 35,000 for each percentage point reduction in power. In the figure above, a full scale comparison is made between standard Becker rudders and Energopac. In this figure results of speed trial measurements are shown for the same series of vessels, where 6 vessels have been equipped with standard Becker rudders and two with Energopac. As can be seen, the power savings vary from 0.5% at low speed up to 4% at maximum vessel speed. In design condition there is a 3.7% reduction in power, resulting in annual savings of more than USD 120,000.



Trial data General Cargo Vessels

Case 2:
A RoPax ferry typically has a high power density propulsion system with additional noise control requirements. This represents an excellent case for considering Energopac. With an installed power of around 25,000 kW and a twin screw CPP installation, Energopac can significantly reduce power requirements. The fuel cost savings can exceed USD 100,000 a year, per percentage point in power reduction.

Case 3:
LNG carriers sail long distances at relatively high speeds. A vessel with a capacity of 135,000 m^3 usually has 28,000 kW installed on a single fixed pitch propeller. Applying Energopac will save roughly USD 135,000 a year in fuel, for each percentage point in power reduction.

Wärtsilä is a global leader in complete lifecycle power solutions for the marine and energy markets. By emphasising technological innovation and total efficiency, Wärtsilä maximises the environmental and economic performance of the vessels and power plants of its customers. Wärtsilä is listed on the NASDAQ OMX Helsinki, Finland.

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